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by

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AN EVALUATION OF THE GASEOUS-STEAM CYCLE
(FIELD CYCLE)

by

Charles A. McPherron

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ABSTRACT

The gaseous-steam cycle or Field cycle is a combination of a Brayton cycle and a Rankine cycle using steam as the working fluid. A portion of the exhaust steam from a regenerative Brayton cycle expands through a condensing turbine. The feedwater is heated by the remainder of the exhaust steam from the Brayton cycle and the resulting wet vapor is compressed to the upper pressure to the Brayton cycle regenerator and heater. The thermal efficiency is higher than either the Brayton or Rankine cycle operating alone. This thesis evaluates the effect of the major parameters of base pressure level, pressure ratio and maximum temperature on thermal efficiency in a cycle that has pressure losses and actual components. The maximum temperature is found to have the greatest effect on thermal efficiency.

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I. INTRODUCTION

The efficient use of energy has become more and more of concern as the cost of fuels increase, fuel reserves are depleted, and the environmental effects of energy by-products are more fully understood. The world's standard of living has become dpeendent on high energy usage. To increase the amount of energy available, two approaches are possible. First, develop new or little-used sources of energy; or, secondly, make better use of present energy sources. New developments are wind, solar and geothermal energy. Efforts toward more efficient use are power plants with higher pressure and temperatures, combined-cycle plants, and fast-breeder-nuclear plants.

At present, the total annual electric-utility power production in the United States is over two billion megawatt hours. This power is produced from nuclear, gas, oil, coal and hydro sources. Figure 1 presents the projected electric utility generation from these sources. It does not seem likely that any new source of power will be developed to take over any significant portion of this production within the future. Thus, conventional electric-power-production methods will continue to be used for many years. The most immediate prospect for increasing energy availability is the improvement of existing power plants. Fossil-fueled-steam plants have efficiencies approaching forty percent, but are not expected to increase much further. Gas-turbine plants have become more important as their thermal efficiencies rise with developments in materials that permit higher operating temperatures. Nuclear power, while producing about ten percent of the total electric power has a high capital cost and environmental opponents. Hydroelectric power production is limited. All of these power production plants have either technological or practical limitations to increases in efficiency. The emphasis on methods of improving plant efficiency has shifted to the inves-

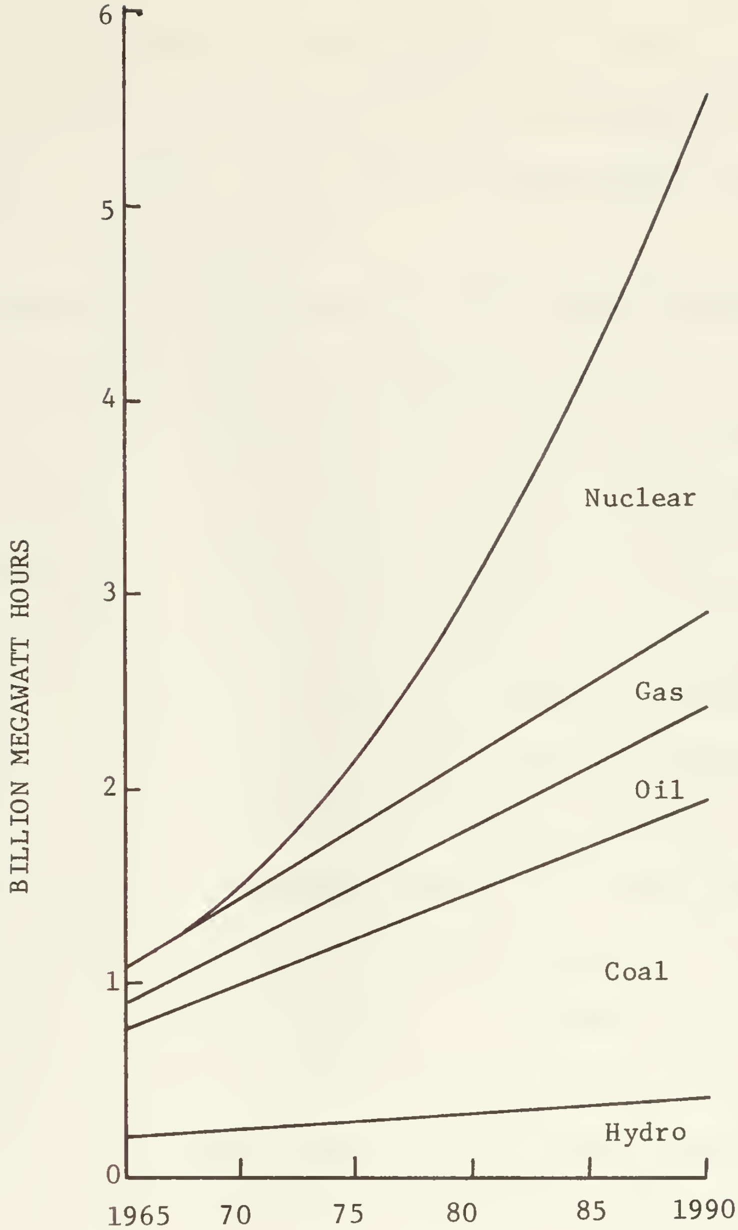

Figure 1. Estimated Annual Electric Utility Generation by Primary Energy Sources [1].

tigation of different types or combinations of thermal cycles that may be capable of higher efficiencies.

One proposal to increase the thermal efficiency of power production is the use of the gaseous-steam cycle or Field cycle, named after J. F. Field who developed it in 1950. This cycle utilizes present technology to make improvements in steam power plants.

The cycle is similar to a regenerative Brayton cycle with steam as the working fluid. The steam is heated to a high super-heated state and expanded through a turbine to a lower super-heated state. The steam then passes through a regenerator where heat is transferred to the high-pressure steam before it enters the heater. The steam flow then divides, part expanding through a condensing turbine and pumped to a spray desuperheater where it is heated by the rest of the steam flow from the regenerator exit. The resulting wet steam is then compressed and flows to the regenerator to receive heat from the exhaust steam from the gaseous turbine.

The advantages of such an arrangement are that the heat addition occurs at high temperatures and heat rejection at low temperatures, thus, in theory approaching the limit of maximum efficiency. The thermal efficiency is high, about forty-five percent, with lower pressures than other steam plants.

This evaluation of the gaseous-steam cycle or Field cycle will first develop and discuss the thermodynamics of the cycle. Secondly, it will determine the sensitivity of cycle parameter variations for a practical cycle. Next, the cycle will be compared to the Brayton and Rankine power cycles on the basis of thermal efficiency and cost. Finally, the future of the Field cycle will be considered.

II. DEVELOPMENT AND DESCRIPTION OF THE GASEOUS-STEAM OR FIELD CYCLE

II.A. Introduction

The gaseous-steam or Field cycle was based on improvements in steam power plants proposed by J. F. Field of the British Electric Authority in 1945. [2, 3, 4] A description of the cycle and its applications was published in 1950 [5]. The purpose of the cycle was to increase the efficient use of power in England. Field's efforts were directed toward improved fuel economy in electric power production and steam powered transportation so that for the same fuel reserves, an increase in power could be used for greater industrial production. The Field cycle was adaptable to electric power production, marine and railway transport and process steam heating. The Field cycle was reported to have a higher thermal efficiency than Rankine cycle steam plants in operation at that time and operate at lower steam pressures. The presentation of the cycle resulted in considerable discussion at that time, but little was done to develop the cycle into an actual power plant. The cycle is more efficient than other power cycles, but improvements in steam-power plants and gas turbines have reduced the efficiency advantage of the untried cycle.

The gaseous-steam cycle has received attention primarily as a textbook example of attempts to achieve high thermal efficiencies and by nuclear-plant designers trying to take advantage of the high-temperature-heat addition and low pressures.

The Field cycle with the superheated-steam turbine called the gaseous turbine, compressor and condensing steam turbine is actually a combination of a Brayton and Rankine cycle using steam as a common working fluid. To show that the cycle is an attempt to achieve maximum efficiency, the Carnot

cycle will be discussed.

II.B. Carnot Cycle

The maximum efficiency that can be obtained is defined by the Carnot cycle. In Figure 2 the processes of the Carnot cycle are 1, heat addition at constant temperature from a heat source; 2, adiabatic-isentropic expansion; 3, heat rejection at constant temperature to a heat sink and 4, adiabatic-isentropic compression to the initial temperature. Although this is an ideal cycle and not obtainable in practice its reversibility demonstrates the highest efficiency possible for a cycle operating between a heat source and heat sink. This efficiency is defined as:

$$\eta = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}$$

where T_1 is the absolute temperature of the heat source and T_2 is the absolute temperature of the heat sink. It is the Carnot efficiency that all thermodynamic cycles attempt to achieve by decreasing the heat sink temperature and increasing the heat source temperature. Most cycles have either a low heat sink or a high heat source but not both.

The Brayton cycle has a high heat addition temperature and a moderate heat rejection temperature. The Rankine cycle for steam plants has a lower heat rejection temperature. If these two cycles could be combined the thermal efficiency could perhaps be higher than either one operating alone.

II.C. Brayton Cycle

A schematic diagram of the Brayton cycle is shown in Figure 3, and the temperature-entropy diagram in Figure 4. Air is compressed from point 1 to 2. Heat is added in the combustor, usually by the combustion of fuel and the products, point 3, are expanded by a turbine to point 4. For an open cycle the exhaust is to the atmosphere. A closed cycle, where the fluid is cycled back through the system, requires a cooler at the turbine

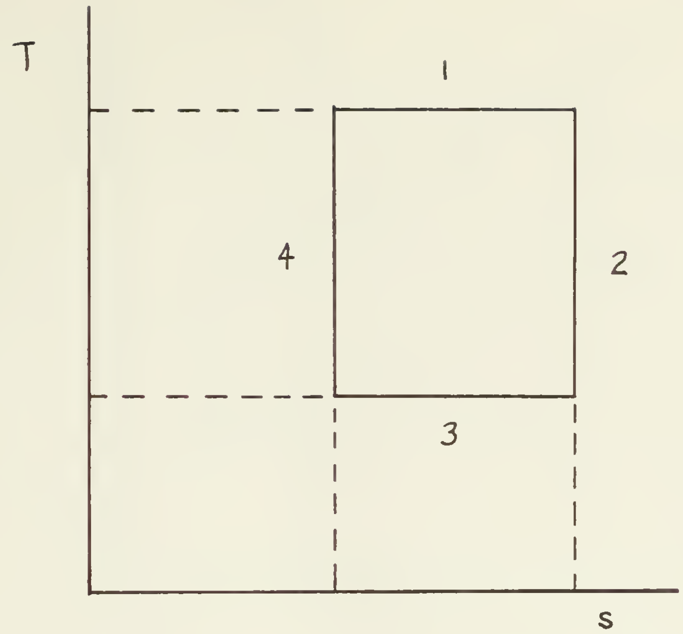

Figure 2. Carnot Cycle Temperature-Entropy Diagram

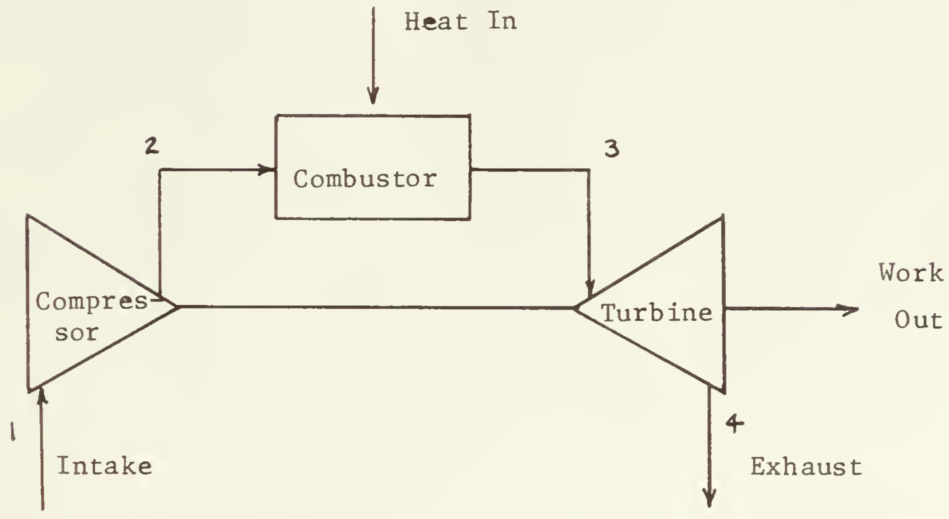

Figure 3. Brayton Simple Cycle

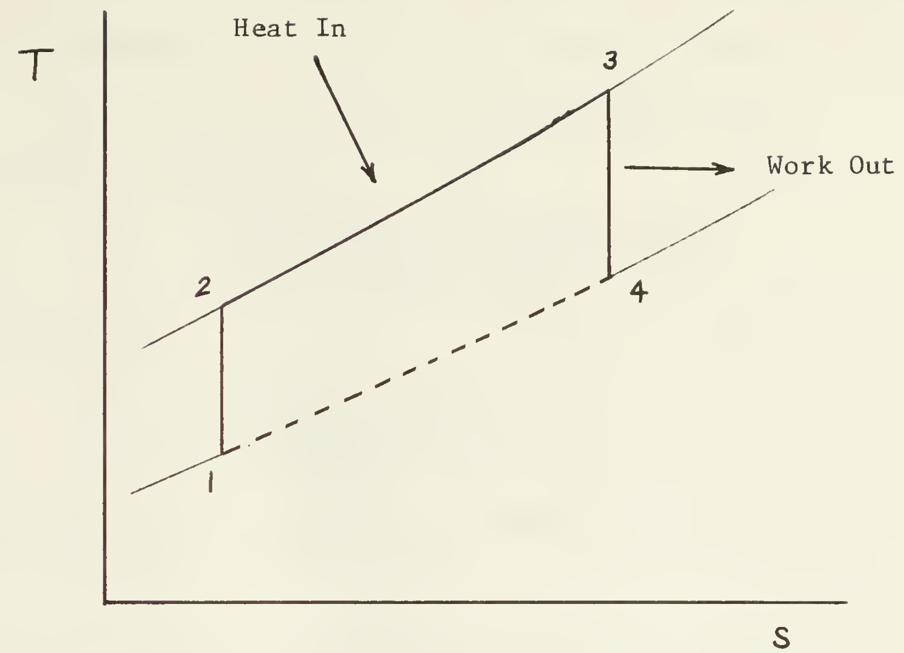

Figure 4. Brayton Cycle Temperature-Entropy Diagram

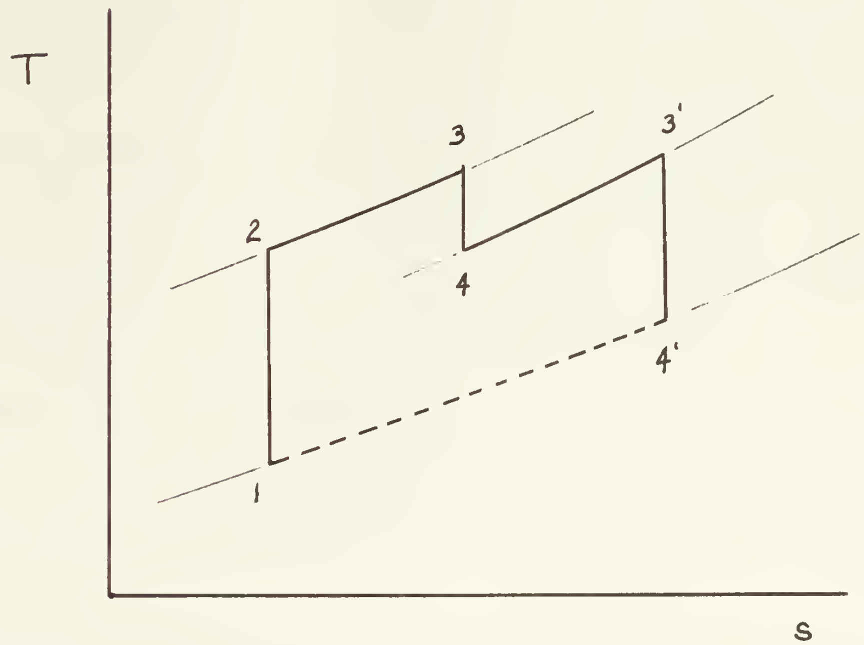

Figure 5. Reheat Brayton Cycle Temperature-Entropy Diagram

exhaust to transfer heat from the exhaust gas and cool the working fluid to point 1. The combustor in a closed cycle becomes a heater with the heat obtained from an external source such as a furnace. The thermal efficiency is defined as:

$$\eta_{th} = \frac{\text{Turbine work - Compressor work}}{\text{Heat input}}$$

or in terms of enthalpy:

$$\eta_{th} = \frac{(h_3-h_4) - (h_2-h_1)}{(h_3-h_2)}$$

neglecting the differences in flow rate in compressor and turbine in the open cycle.

The Brayton cycle can be improved by the following methods:

1. Increased pressure ratio
2. Increased turbine inlet temperature
3. Use of reheat
4. Use of regeneration
5. Use of intercooling

Increased pressure ratio. An increase in the compressor exit pressure moves the heat addition process, point 2 to 3 in Figure 4 to a higher pressure and the increase in turbine work is greater than the increase in the compressor work resulting in an increase in efficiency.

Increased turbine inlet temperature. If the turbine inlet temperature, point 3 in Figure 4 is increased, the increase in turbine work is more than the increase in required heat addition and there is a net gain in efficiency.

Use of reheat. Two turbines are used with the exhaust fluid from the first turbine reheated and expanded through a second turbine. For sufficiently high pressure ratios the net efficiency is increased since the

heat input occurs at a higher average temperature than for one turbine. The temperature-entropy diagram is shown in Figure 5.

Use of regeneration. The use of the turbine exhaust gases or fluid to heat the compressed fluid prior to combustion or heating is called regeneration. An additional heat exchanger is used at the turbine exhaust. The temperature entropy diagram is shown in Figure 6. The exhaust heat is transferred from point 4-4' to the compressed fluid, point 2-2'. The required heat addition is less and the average temperature of the heat addition is higher. The net result is an increase in thermal efficiency.

Use of intercooling. Cooling the fluid between compression stages reduces the work of compression and lowers the average exhaust temperature. This efficiency increase is noted at high compression ratios. The intercooled cycle is shown in Figure 7.

II.D. Rankine Cycle

The Rankine cycle for steam, Figure 8, is the basic cycle for present day, large-scale-steam-power plants. The pump raises water to the boiler pressure at 1, heat is added and the water boils to form saturated steam at 2. The steam is expanded through a turbine to produce work. The turbine exhaust steam, 3, is condensed to water, 4, and again raised to the boiler pressure by the pump. Since the fluid remains in the system this is a closed cycle. To obtain a greater amount of work from the cycle it is customary to make use of a superheater in the boiler to increase the temperature of the steam. This is shown in a temperature-entropy diagram for the Rankine cycle, Figure 9. The labeled points correspond to those of Figure 8. Point 2 represents the saturated boiler conditions. Additional heat in the superheater moves point 2 to 2' in the superheated region. Expansion through the turbine now results in point 3'. The pres-

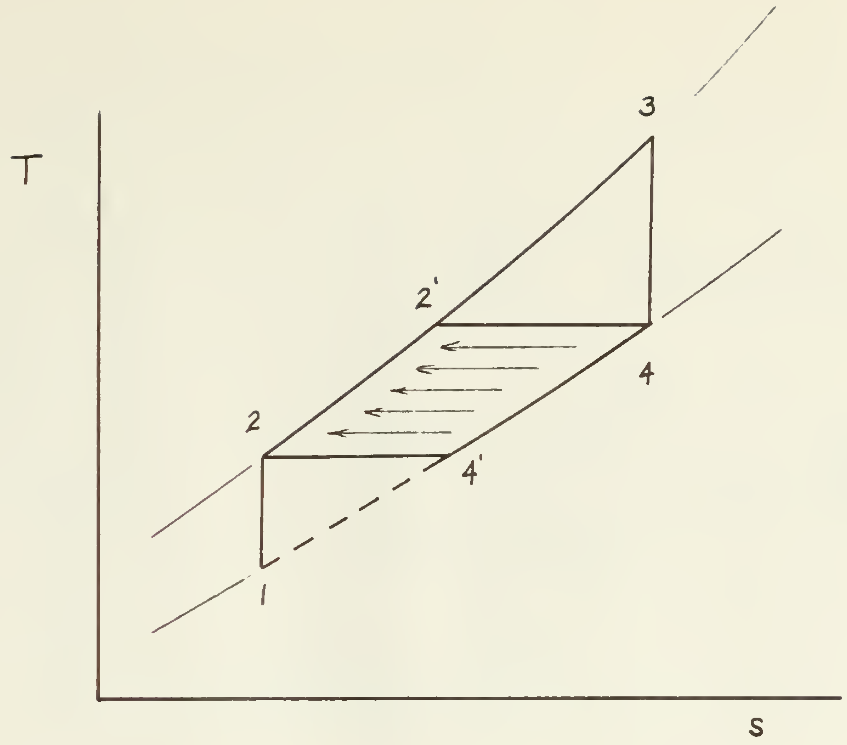

Figure 6. Regenerative Brayton Cycle Temperature-Entropy Diagram

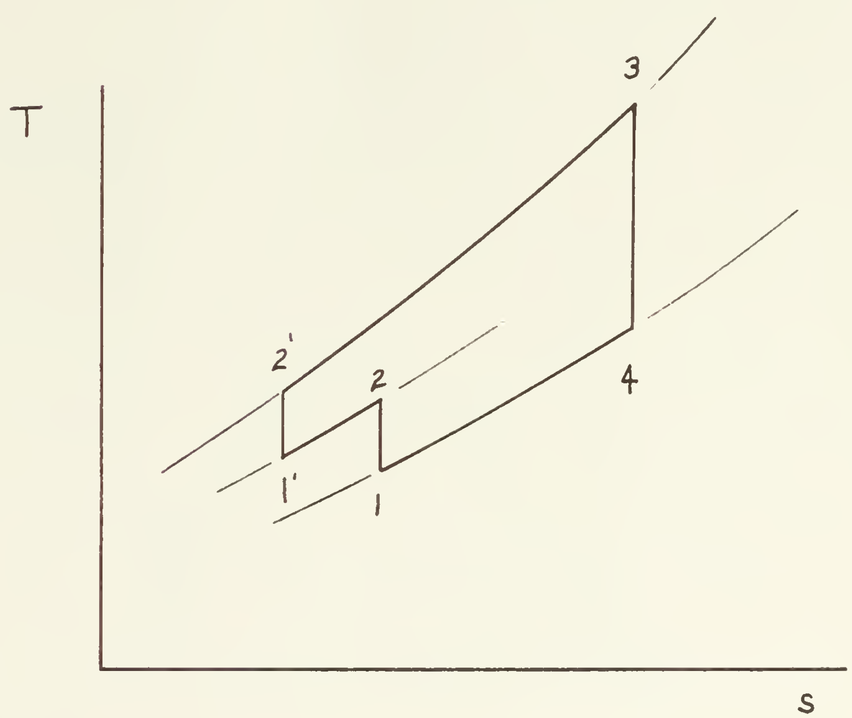

Figure 7. Intercooled Brayton Cycle Temperature Entropy Diagram

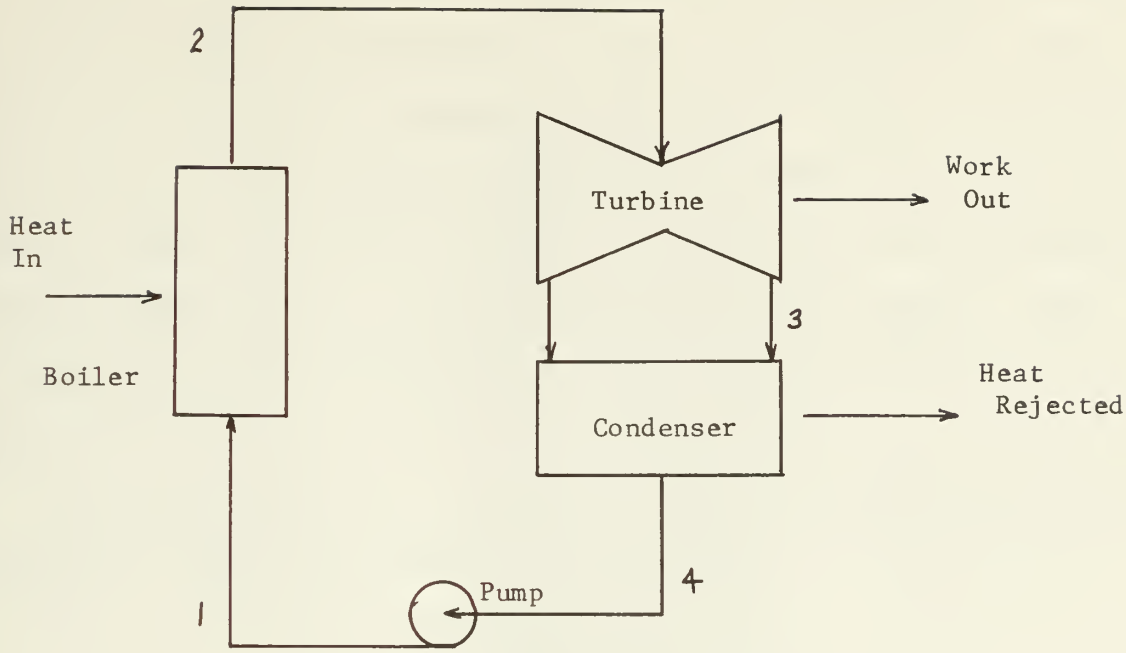

Figure 8. Rankine Steam Cycle Diagram

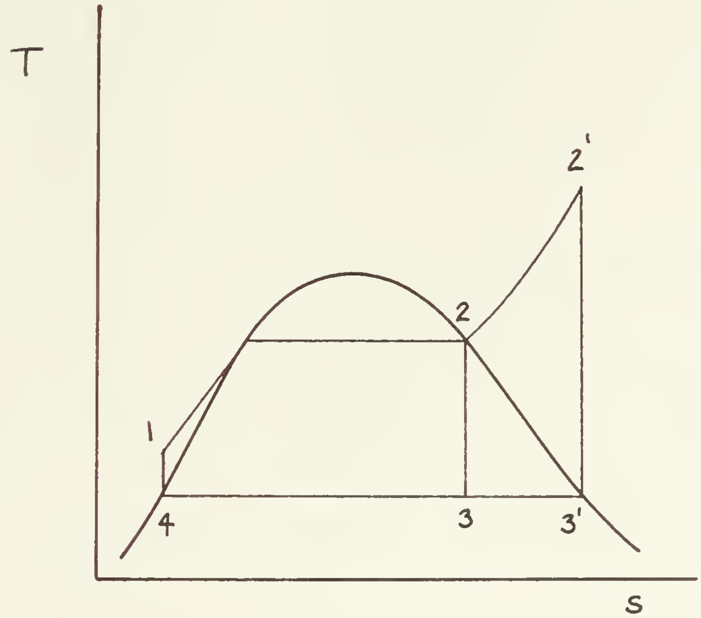

Figure 9. Rankine Cycle Temperature-Entropy Diagram

sure levels are at the same but the temperature difference has increased and more work can be extracted from the cycle. The Rankine cycle described is one of adiabatic isentropic expansion and compression processes. Actual plants, due to practical efficiencies and losses do not have isentropic processes. The points 3 and 3' at the end of the turbine expansion represent water vapor with a certain amount of moisture content. Steam turbines can generally operate with a limited amount of moisture at the exit.

The thermal efficiency of the Rankine cycle can be defined as the net work output of the cycle divided by the heat input:

$$\eta_{th} = \frac{\text{Turbine work - Pump work}}{\text{Heat input}}$$

To improve the efficiency of the Rankine cycle there are five general modifications.

1. Lowering the condenser pressure
2. Increasing the boiler pressure
3. Increasing the superheat temperature
4. Use of reheat
5. Use of feedwater heating

All of these processes have been used in various combinations to improve the efficiency of modern Rankine cycle steam plants.

Lowering the condenser pressure. Since the condenser operates at saturated conditions, reducing the pressure also reduces the temperature. A lower temperature permits a greater expansion through the turbine. This increase in work from the turbine is greater than that required for the increased pump work. The net effect is an increase in efficiency. Modern plants generally operate with a one to two inch Hg vacuum. These low pressures are important to the cycle and are sometimes limited due to the method

of removing the heat.

Increasing the boiler pressure. Figure 10 shows a Rankine cycle with two different boiler pressures. Considering only the superheated case, the turbine work and pump work are about the same for both pressures but the heat input required, 1 to 2 is less for the higher pressure, and efficiency is increased.

Increasing the superheat temperature. An increase in the superheat temperature results in a higher inlet enthalpy to the turbine as well as an increase in the heat addition required in the boiler. For a given plant the percent increase in net work is greater than the percent increase in heat addition and the cycle efficiency is improved.

Use of reheat cycle. The reheat cycle serves to overcome certain practical limitations of steam plants and raise the average heat addition temperature. These limitations are those of the maximum temperature of the steam that can be obtained due to the superheater materials and the avoidance of excessive moisture at the turbine exit. Two turbines are used with the steam from the first being reheated in another superheater in the boiler. The second turbine then can expand to a higher quality. This reheat cycle is shown in Figure 11.

Feedwater heating. To increase the thermal efficiency the heat addition can be reduced by increasing the enthalpy of the feedwater entering the boiler. This can be done as shown in Figure 12 where the steam is taken from the turbine at point 5 and used to heat the feedwater. This decreases the turbine net work since the flow rate is reduced through the latter turbine stages, but less heat is now required to be added due to the higher feedwater enthalpy, and there is a net increase in efficiency.

These are the basic methods used to design modern high capacity steam

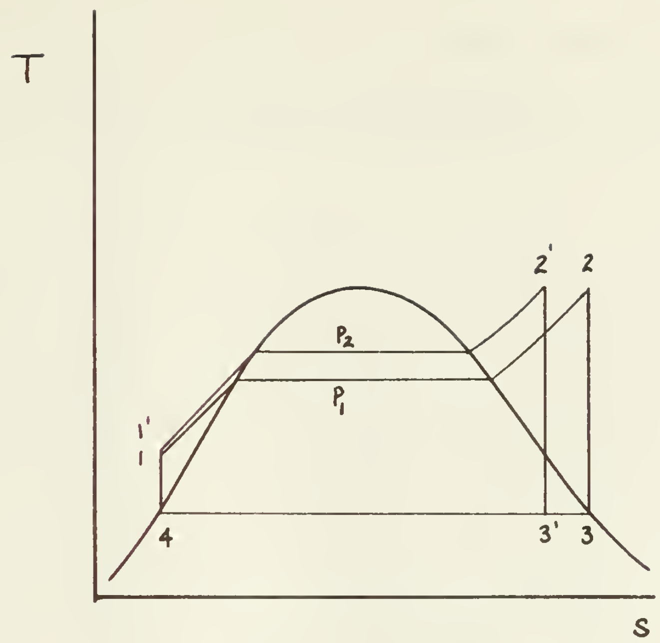

Figure 10. Increased Boiler Pressure Effect on Rankine Cycle

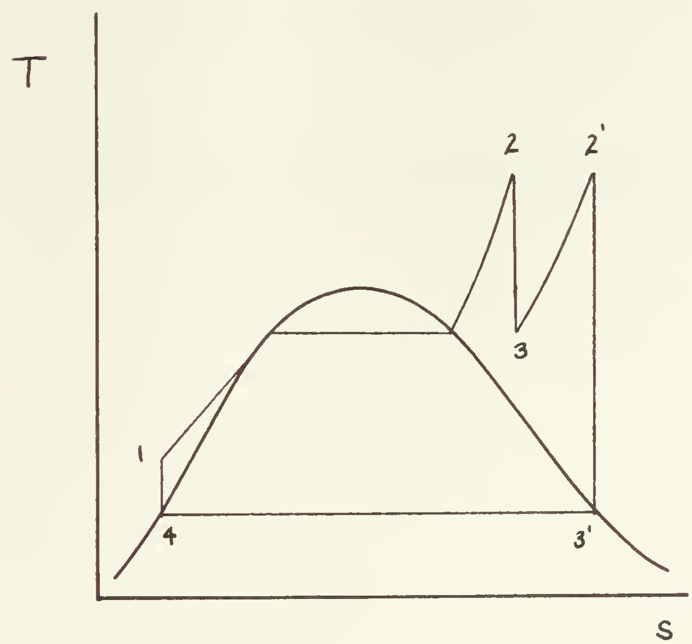

Figure 11. Reheat Rankine Cycle Temperature-Entropy Diagram

power plants. They have been refined through extensive operating experience to obtain station efficiencies approaching forty percent.

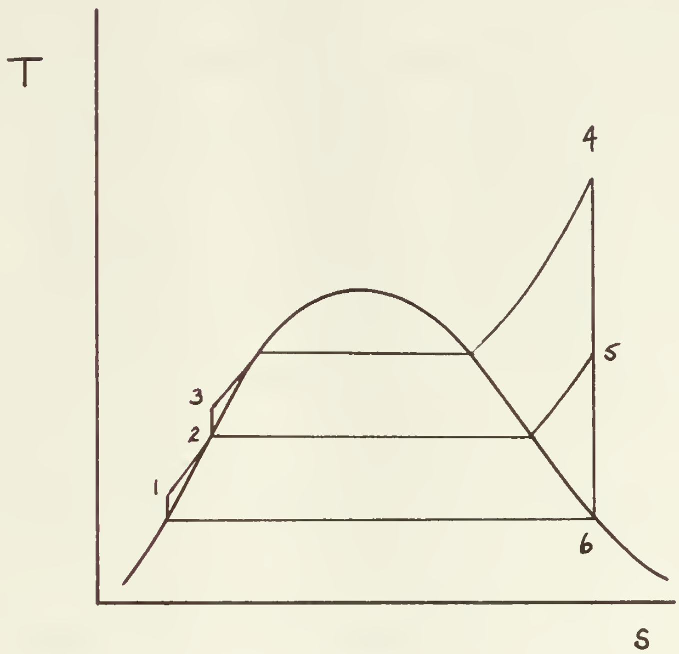

Figure 12. Feedheating Rankine Cycle Temperature-Entropy Diagram

II.E. Ideal Super-regenerative Cycle [6]

The Rankine and Brayton cycles are two of the basic cycles used in the production of power. Each has its advantages and disadvantages. The Rankine cycle operates at low heat rejection temperatures but it is limited by material considerations at the top temperature. The Brayton cycle can achieve higher heat addition temperatures in a gas turbine but also has higher heat rejection temperatures. Combinations of the two cycles might provide higher efficiencies if the high temperature heat addition of the Brayton cycle could be used with the low temperature heat rejection of the Rankine cycle. Furthermore, if the combination of cycles uses steam as the working fluid, there exists no combustion product fouling in the gas turbine or any dealing with liquid metals or pure gases. Steam components required would be similar to present day equipment.

Field proposed such a combination of the Rankine and Brayton cycles over 25 years ago. The cycle known as a gaseous-steam or Field cycle is an attempt to form an ideal super-regenerative cycle. This ideal cycle can theoretically reach the limiting Carnot efficiency based on temperature. A diagram of the ideal cycle is shown in Figure 13 and the temperature-entropy diagram in Figure 14.

Referring to Figure 13, the left hand part of the diagram is the Brayton cycle, the right is the Rankine cycle. In the Brayton cycle superheated steam at point 4 is further heated and expanded in small steps so that the heat addition temperature is nearly constant. The turbines are operating on this superheated steam much like gas turbines. The exhaust superheated steam passes through a regenerator 5 to 6, where it superheats, 3 to 4, the saturated steam coming from compressors. The steam

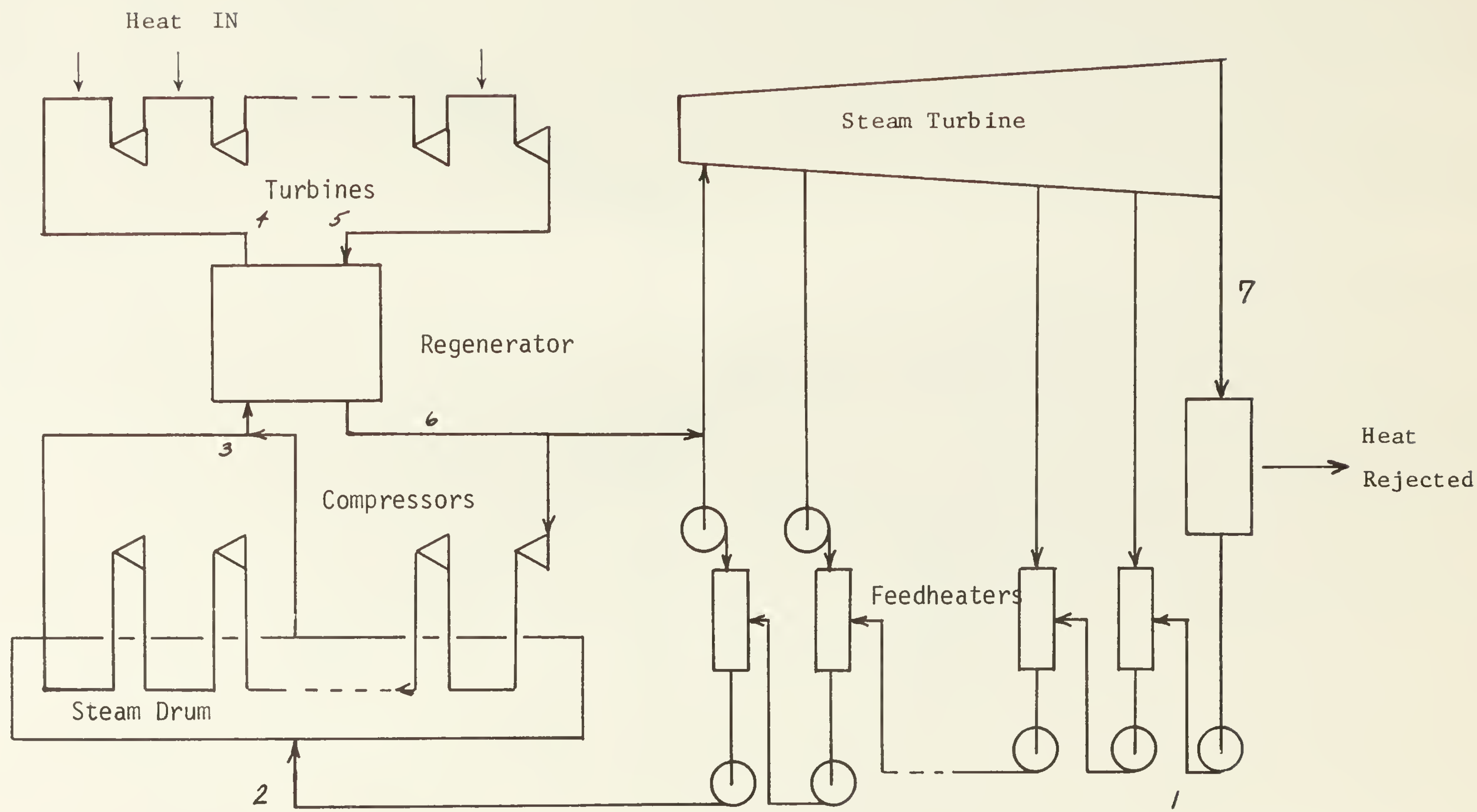

Figure 13. Ideal Super-Regenerative Cycle

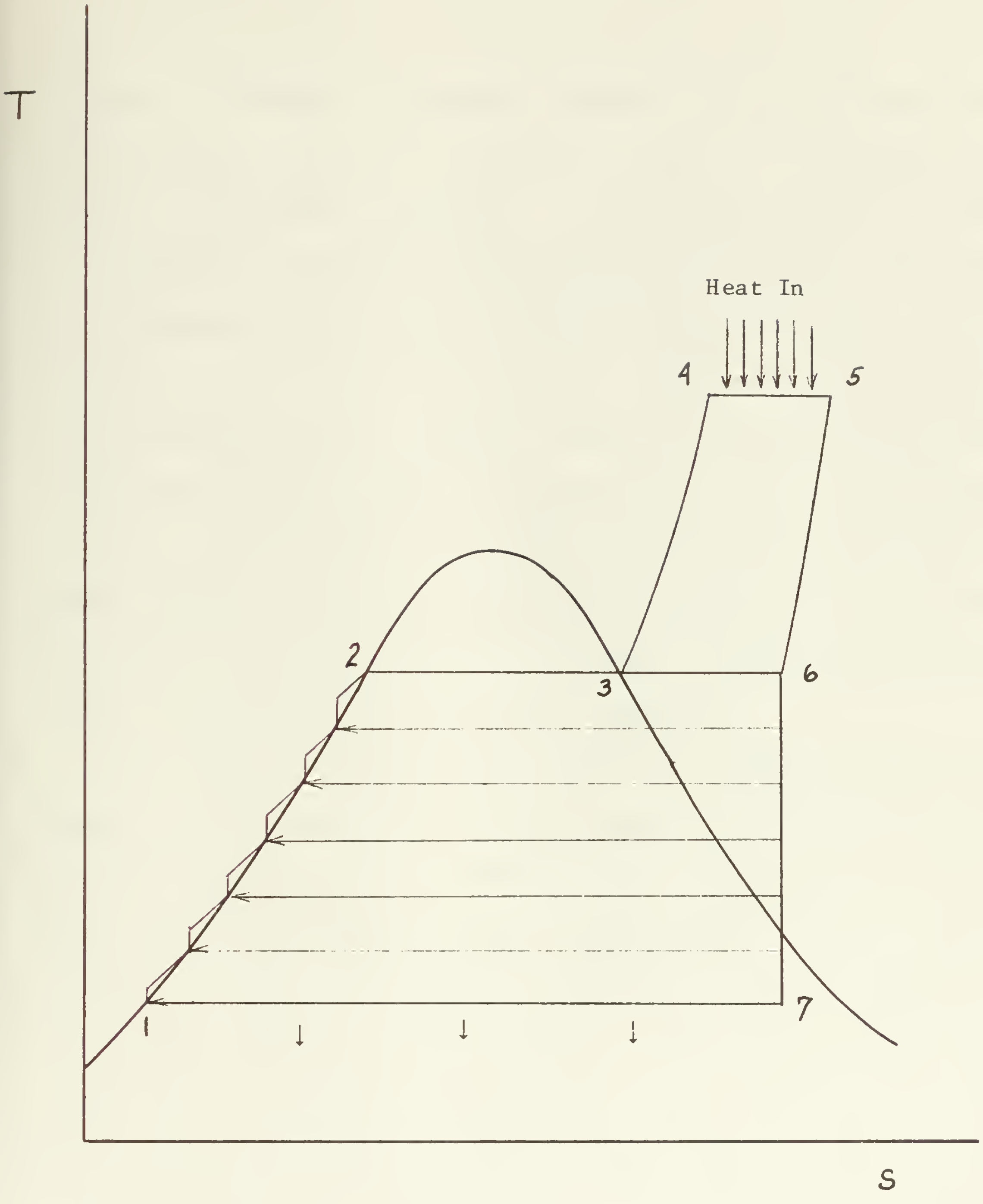

Figure 14. Ideal Super-regenerative Cycle

at point 6, still superheated, is divided, part flows through a regenerative condensing steam plant extracting more work. The exhaust steam from the turbine is condensed at constant temperature and the water is heated in stages to the boiler saturation temperature by steam diverted from the condensing turbine. The feedwater is pumped to a saturated-steam drum. The rest of the steam, from point 6, is compressed to point 3, isothermally and reversibily by multistage compression and intercooling. The heat from the intercoolers evaporates the feedwater in the steam drum to form saturated vapor at point 3 at the maximum pressure. This vapor then is heated, 3 to 4, by the gaseous turbine exhaust. Thus heat is transferred into the cycle only at the top temperature T_2 and rejected only at the low temperature T_1. If there were infinite numbers of compressors, feed heaters, turbines and intercoolers and if steam were a perfect gas, the cycle would be reversible and have the Carnot efficiency. However, this ideal case cannot be achieved so the cycle must be modified.

The changes to the ideal super-regenerative steam cycle are the elimination of the complicated multistage isothermal compression from point 6 to 3 in Figure 13. The reversible mixing is replaced by a spray desuperheater using steam from the gaseous turbine exhaust to heat the feedwater from the condensing steam plant. A compressor is used to compress the resulting wet steam to the upper pressure. The infinite number of reheating stages at the top temperature are reduced to one or eliminated entirely. The resulting Field cycle is shown in Figure 15 and the temperature-entropy diagram in Figure 16. Other features of the cycle remain the same. The gaseous turbine exhaust heats the fluid prior to the heat addition and bleed steam from the condensing turbine is used to heat the feedwater.

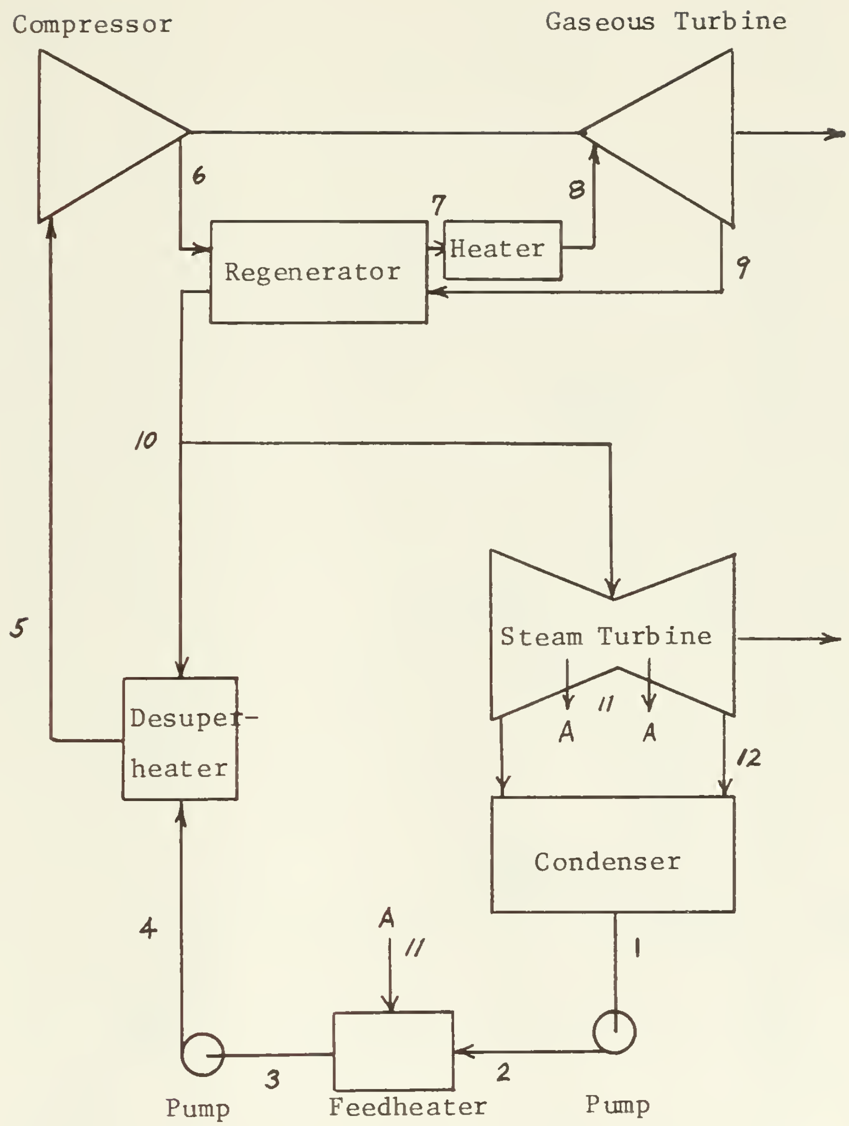

Figure 15. Field Cycle Diagram

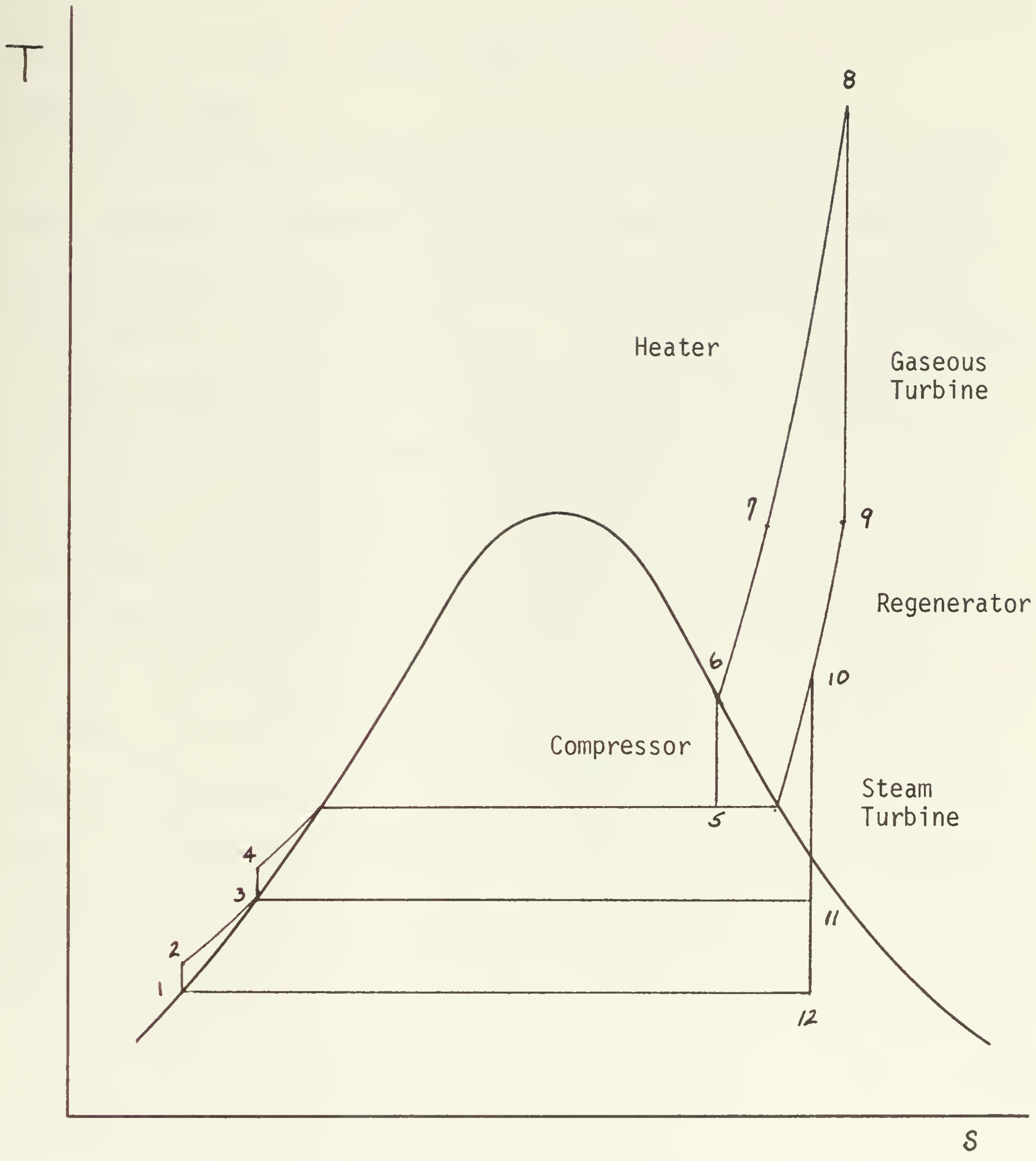

Figure 16. Field Cycle Temperature-Entropy Diagram - Isentropic Components

Thus, there is no heat addition except at high temperatures or rejection except at low temperatures unlike the Rankine cycle that requires heat addition beginning below the boiler saturation temperature or the Brayton cycle that rejects heat at high exhaust temperatures.

II.F. Field Cycle

The Field cycle, developed from the ideal-super-regenerative steam cycle, consists of a Brayton cycle that rejects heat to a Rankine cycle. The Field cycle as presented in Figures 15 and 16, has been discussed only in terms of isentropic processes. To evaluate the cycle in a realistic sense, practical components must be used. The basic parameters of the system are the base pressure level of the spray desuperheater, the pressure ratio of the compressor and the maximum temperature of the heat addition. Once these are set the temperatures in the regenerator and the mass flow through the Rankine cycle become fixed. To illustrate this, the fluid flow through the cycle will be described in detail.

Saturated steam at the compressor exit, point 6 in Figure 16, is heated in the regenerator to 7 where it is further heated by an external source to the maximum temperature at 8. This superheated steam expands through a gaseous turbine to 9 where the still superheated steam is cooled to 10 as it transfers heat to the high pressure steam from the compressor. If the regenerator were perfect and steam an ideal gas, the temperatures at 7 and 9 would be the same as would the temperatures at 6 and 10. This represents the limiting case; the temperature at 7 can be no higher than that at 9 and the temperature at 10 can be no lower than that at 6. For a specific regenerator effectiveness once point 9 is fixed, determined by base pressure, pressure ratio and maximum temperature, points 7 and 10 will be unique.

At the steam turbine inlet, point 10, the flow is divided. About fifteen to twenty percent expands through the turbine to 12 and condensed to point 1. A feedheater is also shown in Figure 16. Feed pumps increase the pressure to point 4 where the feedwater is heated by the remainder of the steam from the Brayton cycle, point 10. The heating occurs in a desuperheater at constant pressure resulting in a wet vapor at 5, the compressor inlet at the base pressure level. The vapor is compressed to saturated steam at the upper pressure where it is heated again in the regenerator.

To determine the mass flow fraction of the steam in the Rankine part of the cycle a heat balance is calculated for the spray desuperheater. The compressed liquid (point 4) is heated by the steam at 10 from the regenerator. A high quality vapor is the result at point 5. If m represents the mass flow fraction through the Rankine cycle then:

$$mh_4 + (1 - m)h_{10} = h_5$$

$$m(h_4 - h_{10}) = h_5 - h_{10}$$

$$m = \frac{h_5 - h_{10}}{h_4 - h_{10}} = \frac{h_{10} - h_5}{h_{10} - h_4}$$

Due to the moderate enthalpy differences for the turbines and the reduced flow through the steam turbine, the flow rate must be high to obtain the same power as other steam plants with similar temperatures. To keep fluid velocities reasonable the components must be large. The low pressures involved also contribute to increasing size. These sizes present design problems and cost considerations.

The components of the Field cycle are for the most part similar to present-day-power plants. The steam turbine is a conventional condensing

turbine operating at low inlet pressures. The condenser and feed pumps are like those of existing steam plants. The gaseous turbine is like a high pressure steam turbine using superheated steam. The regenerator and heater are of the shell and tube type and are combinations of cross and counter flow. Size and cost savings might be obtained by the use of other types of heat exchangers. The compressor operating with wet steam is the most unusual component. It is envisioned as an axial machine with water sprayed into the stages to control heating. The design of such a compressor is recognized as being difficult but design experience in the process chemical industry should be applicable.

To assess whether or not the Field cycle represents a practical method of increasing the efficient use of energy, the cycle must include the losses of an actual power plant. The compressor, turbines and pumps have isentropic efficiencies of less than one. The regenerator effectiveness is also less than one. Pressure losses occur in the piping. The actual Field cycle to be evaluated is shown in Figure 17 and includes efficiencies, regenerator effectiveness and pressure losses. The points as labeled are used throughout the rest of this study.

To determine the efficiency of the cycle quickly a computer program was developed that calculated the thermodynamic properties at each point of the cycle, work output and heat transfer. The computer program is discussed in Appendix A.

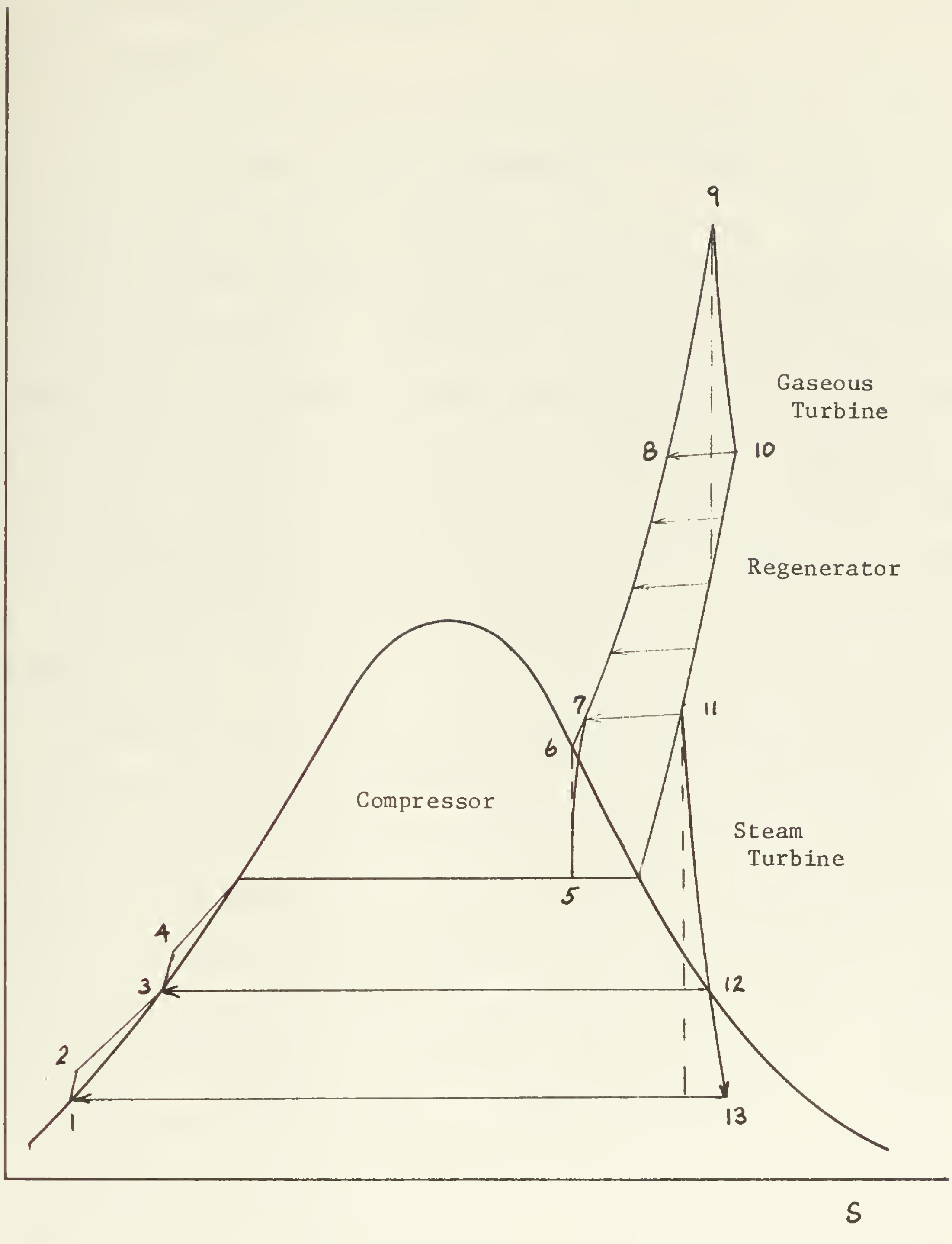

Figure 17. Field Cycle Temperature-Entropy Diagram with Actual Components

III. BASIC FIELD CYCLE PARAMETERS

III.A. General

The actual Field cycle has been described in Chapter II. This chapter will deal with the basic parameters to determine how in a practical sense they affect the efficiency of the Field cycle. The basic parameters are base pressure level, pressure ratio and maximum temperature. Other parameters that are important to the cycle but change little for power cycles are condenser pressure, component efficiencies and pressure losses. In addition, modifications to the cycle such as feedheating and the use of reheat have an effect on efficiency and will be evaluated. Most parameters have limits, either due to the environment of the plant or to the available technology. The parameters that change little are discussed first.

Condenser pressure. The lower the exhaust pressure of a turbine the greater the enthalpy difference. A closed cycle permits exhaust pressures below atmospheric to be used, lowering the heat rejection temperature with a theoretical increase in efficiency. Modern steam plants have condensers operating at from one to two inches of Hg vacuum. These low vacuums are possible only if a sufficient volume of low temperature cooling fluid is available. Generally river or lake water is used for cooling and these vacuums are used. If dry-air-cooling towers are necessary the condenser pressure must be increased since the air temperature is higher than that of river or lake water. One environmental concern is the amount of heat rejected for large power plants. This heat, called thermal pollution, is often sufficient to affect the ecology of the cooling river or lake. It is an advantage of the Field cycle that less heat is rejected

for the power obtained in comparison to other plants due to the lower mass flow rate through the condensing cycle. For this study a condenser pressure of one psia has been selected as a representative value.

Component efficiencies. The performance of the systems components are important influences on efficiency. Field received much criticism for his choice in 1950 of a gas turbine efficiency of 91%, a steam turbine efficiency of 85%, compressor efficiency of 85% and a regenerator effectiveness of 91%. Perhaps for that time they were high but today they do not seem so unreasonable particularly in regard to large scale power plants. Condensing turbines of 500 MW or larger have efficiencies of near 85%. The non-condensing gaseous turbine operating at high inlet conditions and only in the superheat region should be able to reach efficiencies of 90%. The compressor will be assumed to have an efficiency of 90%. The effectiveness of the regenerator is taken as .9. Pump efficiencies are set at 80%.

Pressure Losses. In addition to component efficiencies and regenerator effectiveness the only other major parameter that has a general limit is the pressure loss. Losses occur throughout the system but the most important pressure losses occur in the regenerator and heater. A total of a five percent pressure loss has been chosen. Half of the loss is taken to occur on the high pressure side of the regenerator. To standardize and simplify the calculations the compressor exit pressure is fixed at the high pressure and pressure losses occur up to the constant maximum temperature. The regenerator low pressure inlet side pressure was fixed at two and one-half percent above the basic low pressure so that the regenerator low pressure exit pressure is the base pressure. To illustrate

the change in the cycle due to pressure losses Figure 18 plots the differences in efficiency for the cycle with and without the five percent pressure loss. The base pressure is 100 psia, the maximum temperature is 1200^{o}F. The pressure loss causes a loss of efficiency of about one and one-half percent.

Feed heating. A feature of modern steam plants is the use of feed heating to raise the enthalpy of the feedwater and thus reduce the heat input. In the Field cycle heat addition occurs only in the upper part of the cycle in the superheat region. The feed heating is carried out in the spray desuperheater. Thus, feed heating in the Field cycle does not reduce the required heat input. To investigate feed heating in the cycle a single feed heater was placed in the condensing part of the cycle. The results are plotted in Figure 19. The efficiency increases are due to the increased mass flow through the condensing turbine. The increase is about one-half percent. The feed heater was set to operate at a temperature half-way between the condensing temperature and the low pressure saturation temperature. The condenser pressure, component efficiencies, pressure losses and a single feedheater have thus been included in the Field cycle. These parameters will not be changed while the base pressure, pressure ratio and maximum temperature are varied. The thermal efficiencies are about forty-five percent for the base pressure of 100 psia and pressure ratios of 3:1 and 4:1.

III.B. Base Pressure Level and Pressure Ratio.

The Field cycle is more like a Brayton cycle than a Rankine cycle in that it has a base pressure and a pressure ratio. It is considered an advantage of the Field cycle that its efficiency is high without the high

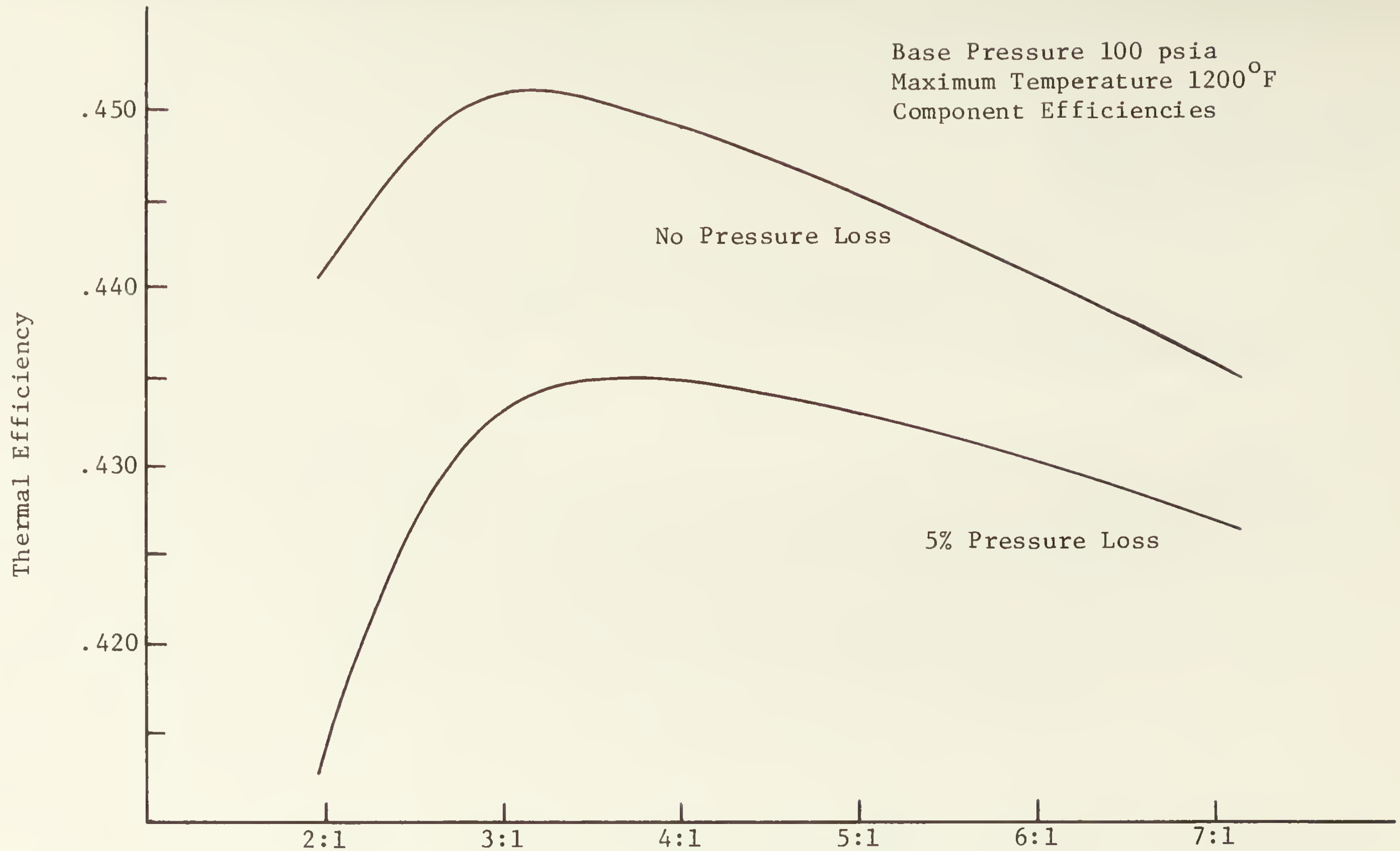

Figure 18. Efficiency vs. Pressure Ratio - Pressure Loss

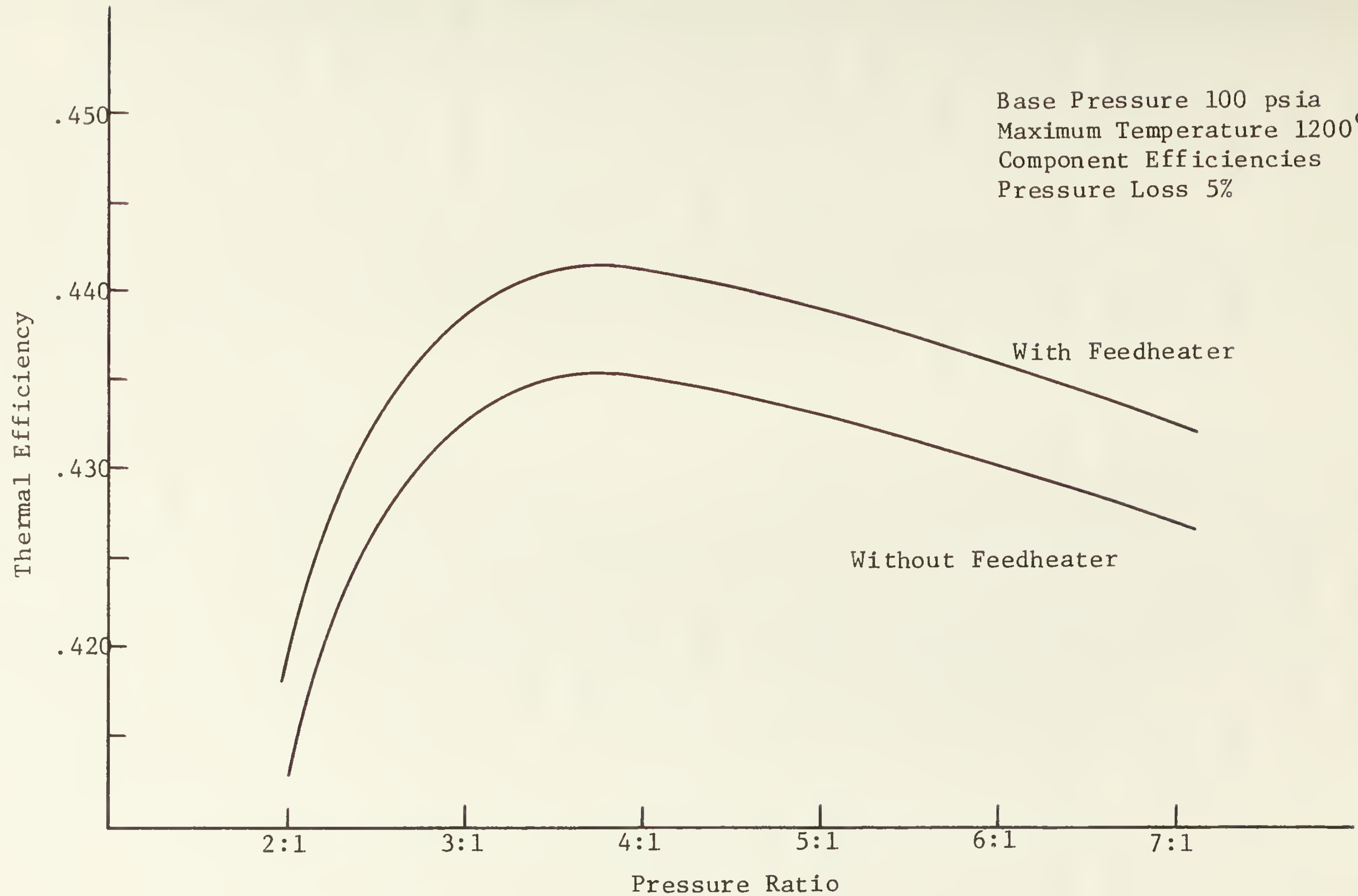

Figure 19. Efficiency vs. Pressure Ratio - Feedheating

pressure requirements of other steam plants. To evaluate the changes due to pressure level and pressure ratio, base pressures of 100, 200, 300, 400 and 500 psia were used in the computer program with pressure ratios of two to seven. The results indicate that as the base pressure level increases, efficiency also increases. Thermal efficiency and pressure ratio at the selected pressure levels are presented in Figure 20. The base pressures of 100 and 200 psia have a maximum efficiency between pressure ratios of 3:1 and 4:1. Base pressures of 400 and 500 psia have increasing efficiencies with no defined maximum. The efficiencies of the base pressures of 300, 400 and 500 psia are not evaluated at higher pressure ratios because the gaseous turbine exit temperature approaches the compressor exit temperature resulting in no regeneration. The increase in efficiency with decreasing regeneration is due to the increase in specific heat with pressure for steam. The decrease in regeneration and the increase in enthalpy changes for the turbines is illustrated in Figure 21. Here the base pressure is 500 psia, the maximum temperature 1200^{o}F. As the pressure ratio is increased, the gaseous turbine exit enthalpy decreases, the compressor exit enthalpy increases and this reduces the enthalpy difference available between the inlet and outlet of the regenerator. At a sufficiently high pressure ratio the gaseous-turbine-exit temperature is equal to the compressor exit temperature so that no regeneration is possible. There can be no further increase in pressure ratio since the gaseous-turbine-exit enthalpy and steam-turbine-inlet enthalpy will coincide.

The steam turbine exit condition consists of a higher than usual moisture content. At a 4:1 pressure ratio for the base pressure of 500 psia the moisture content is 14.8 percent. This is higher than the eight to twelve percent normally considered for condensing steam turbines. Special

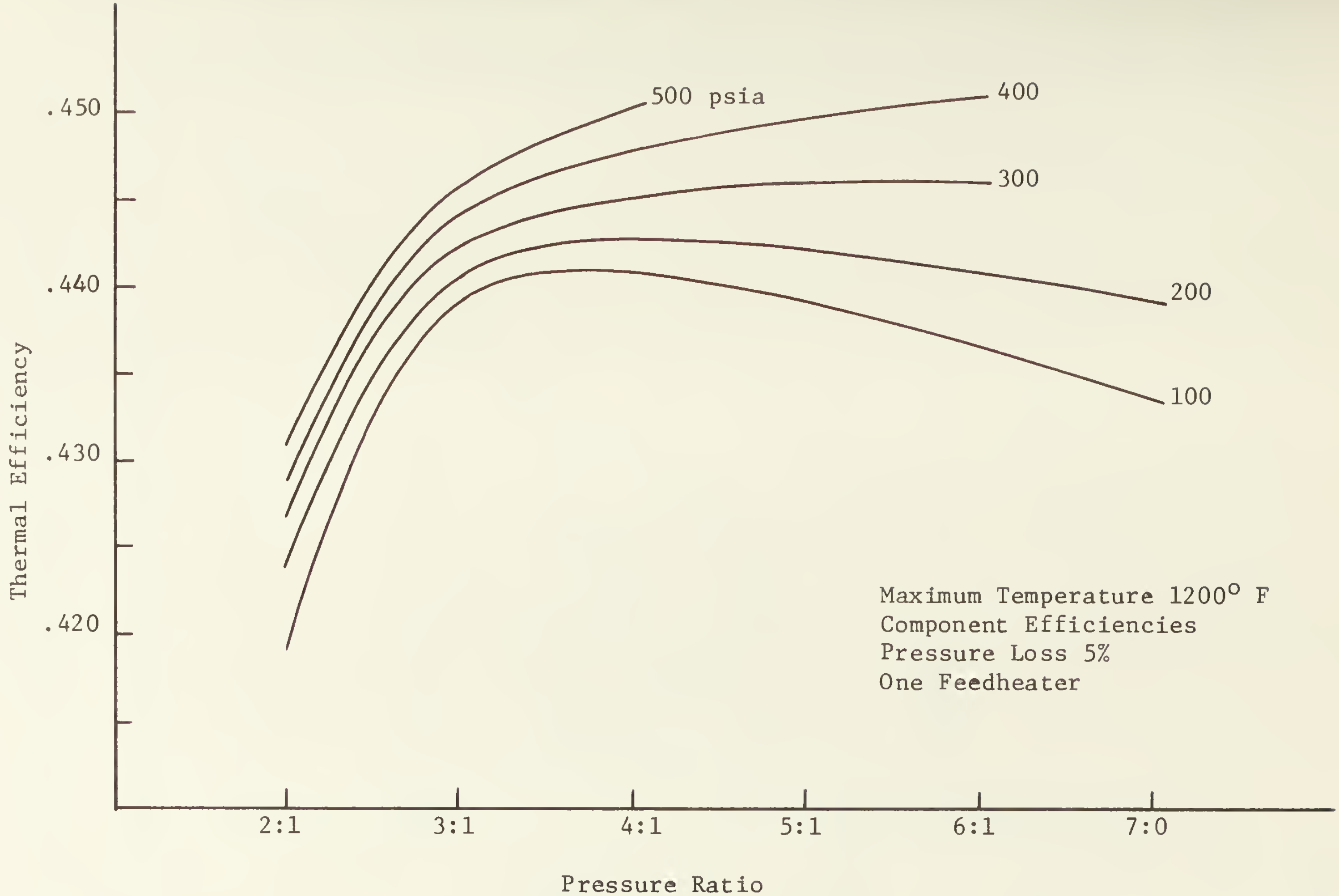

Figure 20. Efficiency vs. Pressure Ratio - Base Pressure Levels

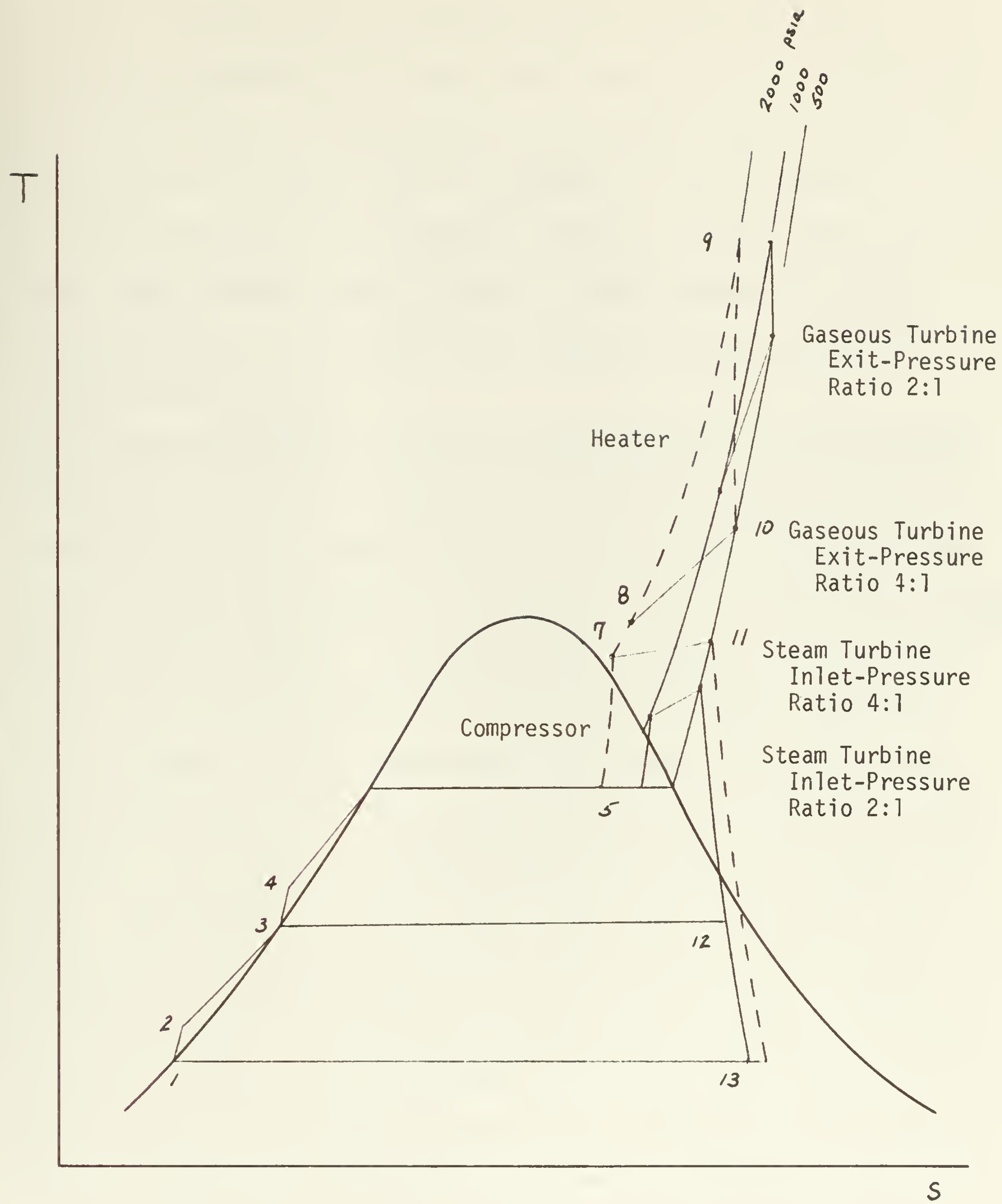

Figure 21. Temperature-Entropy Diagram for Base Pressure of 500 psia

moisture extraction stages can reduce this by one or two percent or reheat gaseous turbines may be used. Additionally, the maximum pressure of 2000 psia is not in keeping with the low pressure, high efficiency attractiveness of the Field cycle. Therefore, for base pressures of 300, 400 and 500 psia the high moisture content and high pressures tend to make these pressures levels unuseable. A base pressure of 200 psia and a pressure ratio of 4:1 with a twelve percent moisture content is the more realistic cycle.

III.C. Maximum Temperature. Thus far 1200°F has been used as the maximum temperature for the cycle. This has been used for Field cycle investigations since Field first proposed the cycle. It was based on the expected increase in steam temperatures in the 1950's. However, the expected increases did not occur in practice. Modern power plants operate at 1000 or 1050°F because of the metallurgical limits of materials for fossil fuel plants. The few attempts at higher temperatures have not been successful, namely the Eddystone plant of the Philadelphia Electric Company that was designed for 1200°F. More will be said later about upper limits but assuming that higher temperatures are possible, what is the effect on the Field cycle? To determine this, temperatures of 1200, 1400, 1600 and 1800°F were examined in the cycle. The efficiency as expected increased with increasing temperature for the given base pressure level of 100 psia. This plot is Figure 22 and the temperature-entropy diagram is Figure 23. The efficiencies show significant increases with increases in temperature. Going from 1200°F to 1400°F adds four percent efficiency. This temperature level increase is thus, the most important parameter that influences efficiency. The moisture content of the steam decreases with increasing temperature.

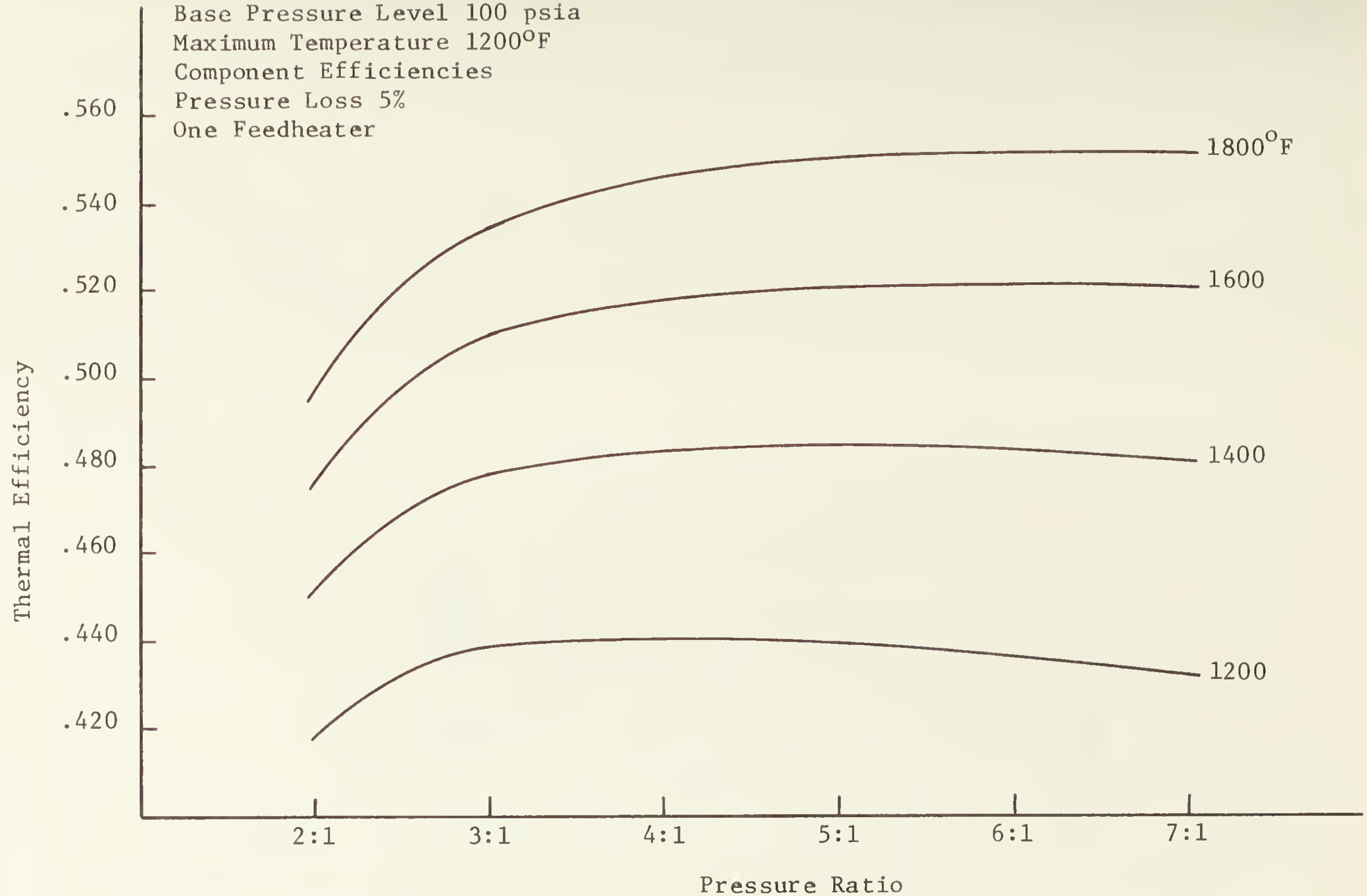

Figure 22. Efficiency vs. Pressure Ratio - Maximum Temperature-Base Pressure Level 100 psia.

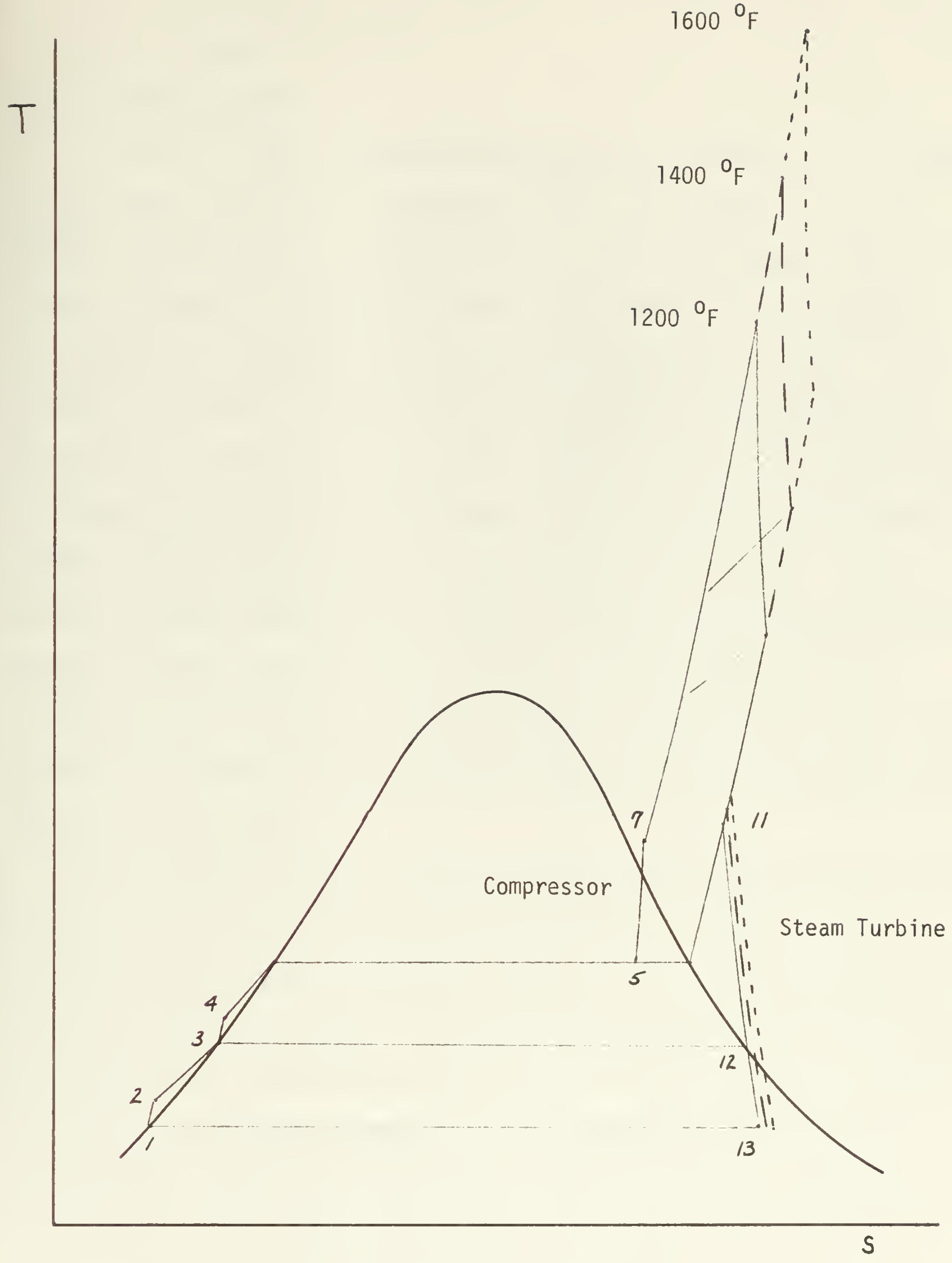

Figure 23. Temperature-Entropy Diagram for Increasing Maximum Temperature

Temperature increases for a base pressure of 200 psia resulted in similar efficiency increases, Figure 24.

III.D. Reheat Turbine

It has been noted in previous sections that the quality of the steam at the steam turbine exit is as high as 15%. One method to reduce the moisture content is to use reheat turbines. This cycle modification is shown in Figure 25. The reheat pressure was chosen as midway between the high and low pressures with the maximum temperature at 1200^{o}F. The results are plotted in Figure 26 for base pressures of 100 and 500 psia. The increase in efficiency for the base pressure of 100 psia is about two percentage points and for the 500 psia base pressure is one and one-half points. Reheat permits greater pressure ratios to be attained. This is due to the higher gaseous turbine exhaust conditions of the second turbine which are higher than for a single turbine. Regeneration begins at a greater enthalpy and higher pressure ratios are reached before the gaseous turbine exit temperature approaches the compressor exit temperature and no regeneration is possible.

The thermal efficiency of the Field cycle has been evaluated for the effects of base pressure and pressure ratio, maximum temperature and reheat for fixed condenser pressure, component efficiencies, pressure losses and for a single feed heater. The cycle has been shown to have efficiencies of 44-45 percent at pressure levels of 100 to 500 psia and pressure ratios of 4 to 6. If a moisture content of twelve percent is considered as the maximum for condensing turbines the cycle is limited to pressure levels of 100 psia at all pressure ratios, 200 psia at ratios of 4:1 or more, and 300 psia at 6:1. The maximum efficiency is 44.5 percent for a base pressure of 300 psia and a pressure ratio of 6:1. If high pressures are

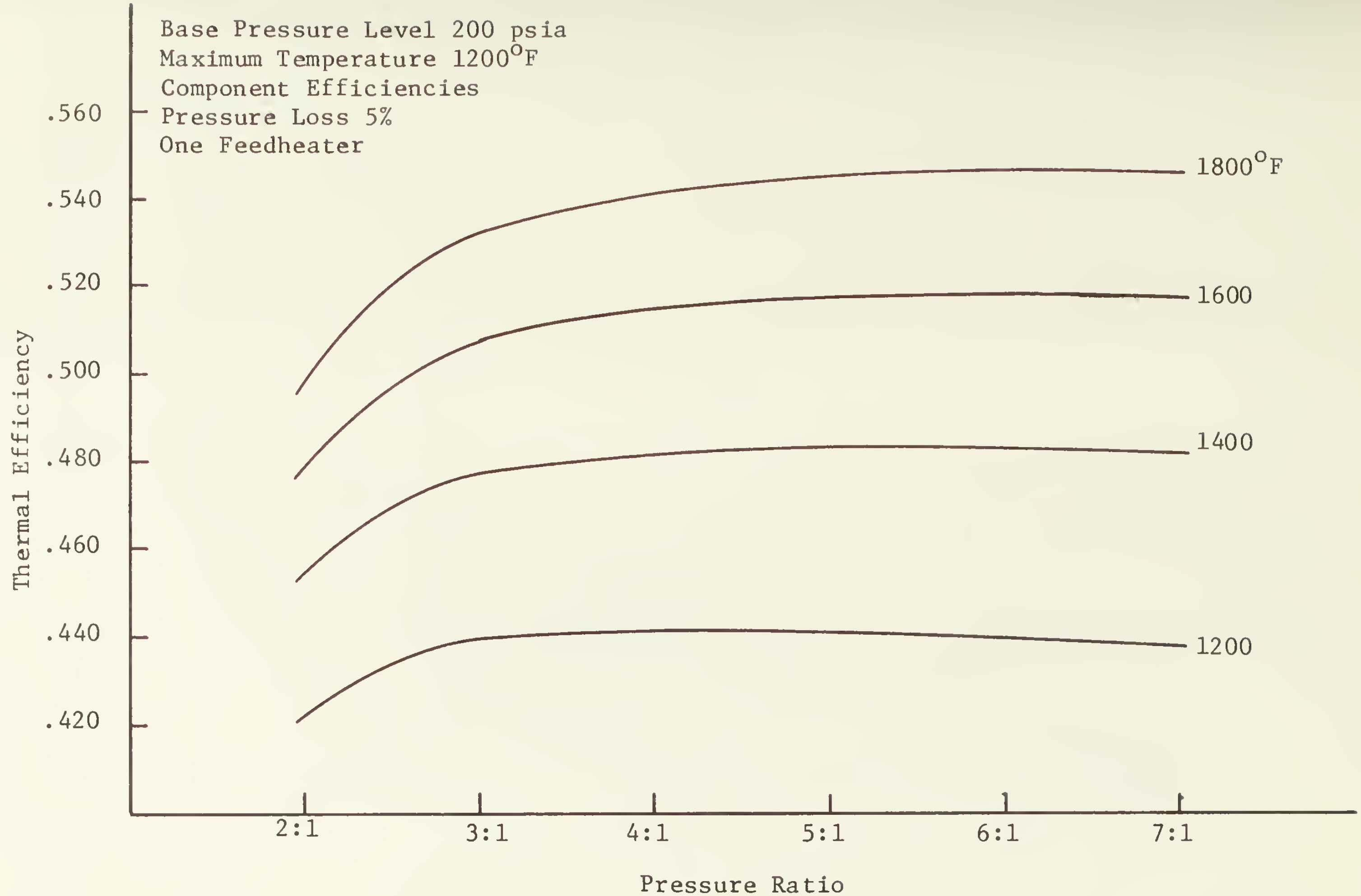

Figure 24. Efficiency vs. Pressure Ratio - Maximum Temperature-Base Pressure Level 200 psia.

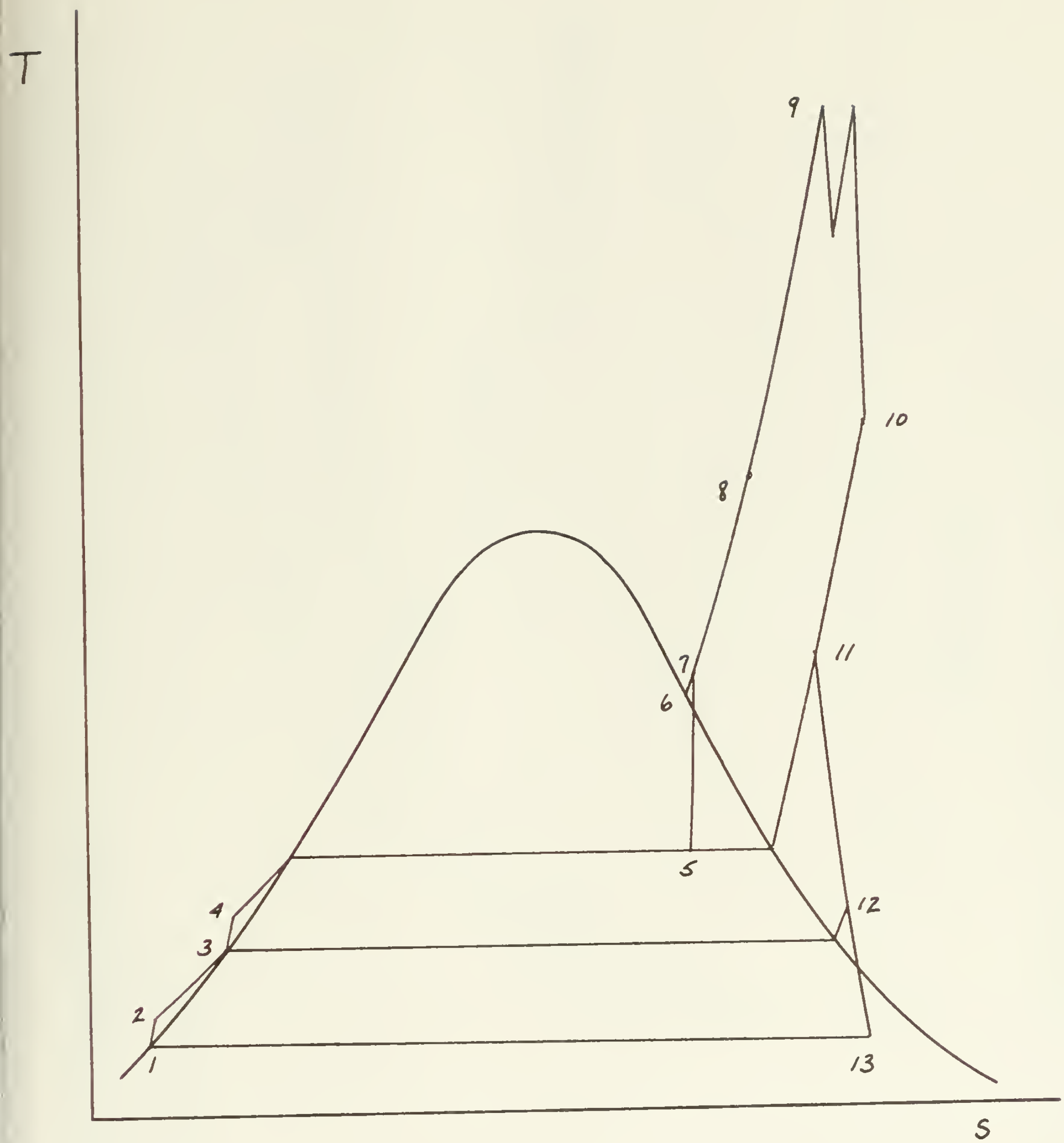

Figure 25. Reheat Field Cycle Temperature-Entropy Diagram.

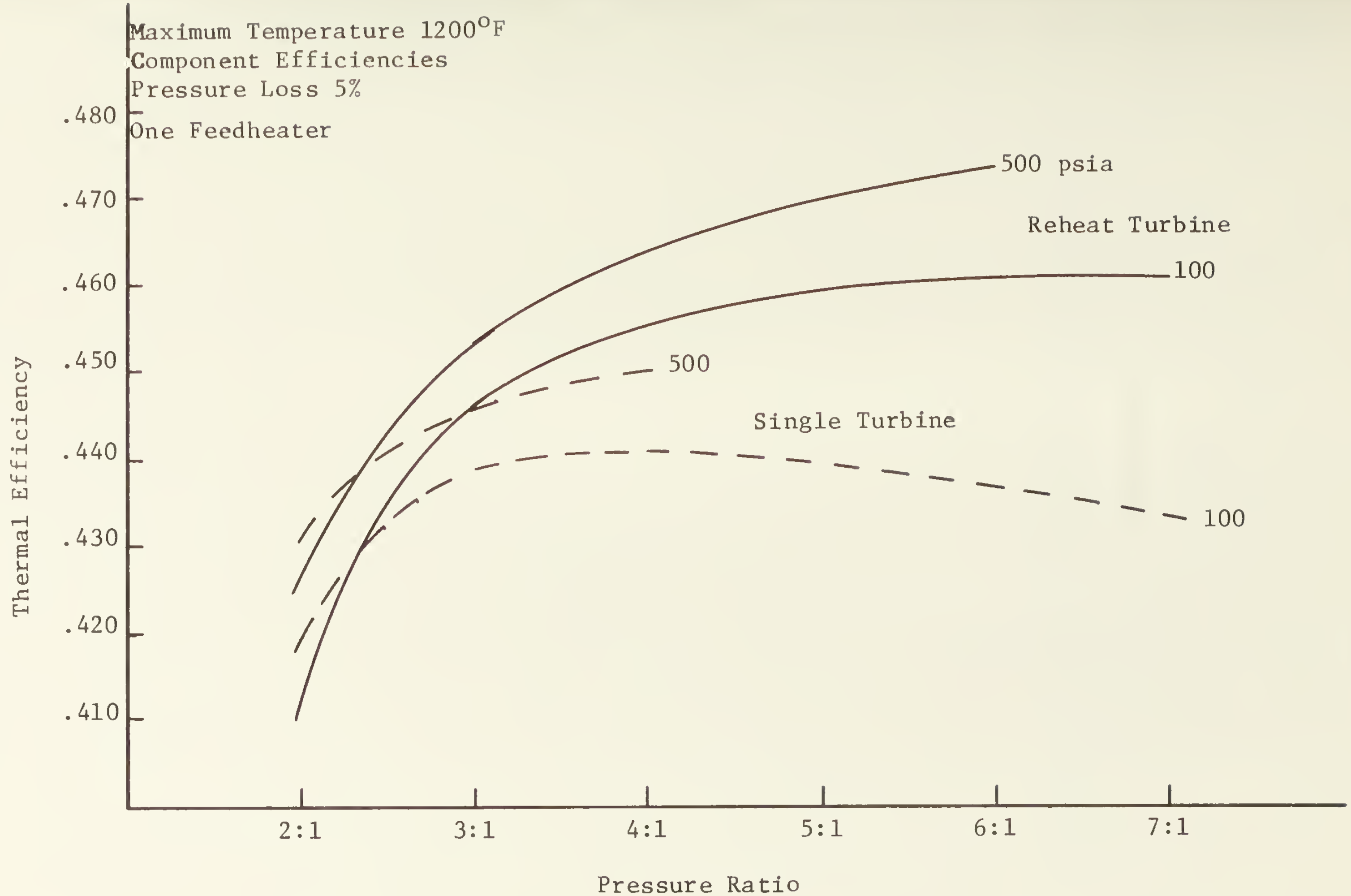

Figure 26. Efficiency vs. Pressure Ratio - Reheat Turbines

to be avoided the next best pressure is 200 psia at a ratio of 4:1.

Reheat cycles reduce the moisture content by only about .4 percent. The pressure levels of 400 and 500 psia would still have too high a moisture content. Base pressures of 300 psia at a ratio of 6:1 or 7:1 and pressures of 200 psia with ratios of 3:1 or greater, and 100 psia would all have 12 percent or less moisture. Efficiencies for the reheat cycles are from 46 to 47 percent at pressure ratios of 5 or greater.

IV. BRAYTON AND RANKINE CYCLE COMPARISON

The value of the Field cycle as a method of electric power production is its high thermal efficiency at lower pressures than those in conventional steam plants. This chapter will provide a comparison of a Field-cycle power plant with steam and gas-turbine plants. The Brayton-gas-turbine plant is taken from a report by United Aircraft Research Laboratories [7]. The plant is a 1980's design of a regenerative gas turbine with a turbine-inlet-temperature of 2400°F and a compression ratio of 12:1. The net plant efficiency is 37.6 percent. The Field cycle plant would be similar to the gas turbine plant in that it has a compressor, non-condensing turbine and a regenerator. The Field cycle has heat addition in a heater while the gas turbine uses a combustion unit. The differences are that the gas turbine exhausts to the atmosphere after the regenerator while part of the exhaust from the regenerator in the Field cycle is used in a condensing turbine to increase the work output from the cycle. This results in a lower heat rejection temperature for the Field cycle.

The Rankine steam plant is also from the same United Aircraft report [7] and represents similar technology. The pressure is 3500 psia with a 1000°F reheat turbine. The net plant efficiency is 38 percent. The Field plant is similar in that it also uses a condensing turbine, condenser and feed pumps. The differences are that in the Field cycle, the feedwater is heated by the major part of the superheated steam coming from the regenerator and the wet vapor is compressed to the heater pressure where in the Rankine cycle the feed pumps increase the pressure to that of the boiler.

To compare the plants a heater efficiency of 90 percent is assumed for the Field plant. The three plants are summarized in Table 1.

Table 1. Brayton, Rankine and Field Cycle Utility Plant Comparison. 1000 Megawatts.

	Brayton Gas Turbine	Rankine Steam	Field Steam
Characteristics	$2400^oF/12:1$ Regenerative	3500 psia 1000^o reheat	200/800 psia 1200^oF
Net Efficiency, %	37.6	38.0	39.8
Heat Rate, BTU/KWHR	9075	8980	8580

Based on the values in Table 1, the Field cycle has a small advan- in thermal efficiency that will result in lower fuel costs. The Field plant operates at lower pressures than the Rankine plant and thus will have a high volume flow rate which means large piping and components. The enthalpy differences in the Field plant are small resulting in a high mass flow rate that will also increase component size if acceptable fluid velocities are to be maintained.

The only apparent way to increase the thermal efficiency of the Field cycle so that it has a significant increase in efficiency over the other plants is to raise the maximum temperature. It has been shown that thermal efficiencies of 48 percent are possible for temperatures of 1400^oF. If reheat turbines are used, the thermal efficiency should approach 50 percent with a plant efficiency of 45 percent.

V. COST COMPARISON OF BRAYTON, RANKINE AND FIELD CYCLE UTILITY PLANTS

The Field cycle has a higher thermal efficiency than either a gas turbine or a conventional steam electric power plant described in a recent study [7]. This higher efficiency will result in lower operating fuel costs for the Field cycle over a conventional steam plant if both use the same fuel and may be lower than the Brayton cycle gas turbine which can use low cost gas as a fuel. The purpose of this chapter is to make a preliminary estimate of the fuel and capital costs of Brayton, Rankine and Field cycle electric utility plants. The cost estimation method will be to consider representative utility plants from reference 7, and use average values for fuel and equipment costs. It is recognized that such an approach does not take into account the wide variation of fuel costs throughout the United States, but the intent is to determine if the cost of the Field cycle utility plant is on the same order of magnitude as Brayton and Rankine cycle plants. If the plant costs are comparable, detail designs could then be made. The fuel costs will be compared first and then the capital costs.

V. A. Fuel Costs

Fuel costs, Table 2, were obtained from a 1973 report to the Energy Policy Project of the Ford Foundation [8]. Since Eastern United States utilities use coal primarily and little gas, coal burning Rankine and Field plants will be compared to an oil burning Brayton plant. Also, all three plants using gas such as in the Southwest will be compared. The efficiencies are those from Chapter IV for the 1980's designs. The fuel costs per year for the three plants are listed in Table 3. The Field cycle because of its lower heat rate, is competitive with other plants if the type of fuel and the region of the United States are considered. This lower fuel cost of

the Field cycle means that the capital cost can be higher by the amount of the fuel savings over the years.

TABLE 2. FUEL PRICES [8]

Fuel	Average Price	
Coal	$\$.69/10^6$	BTU
Oil	.90	Distillate
	.81	Residual
Gas	.26	(Western United States only)

TABLE 5. FUEL COSTS PER YEAR FOR UTILITY PLANTS

1000 MW

70% Capacity

1973 Dollars

Plant	Fuel	Plant Efficiency %	Plant Heat Rate BTU/KWHR	Fuel Cost Year $\$ \times 10^6$ Eastern United States	Fuel Cost Year $\$ \times 10^6$ Western United States
Brayton-Gas Turbine	Oil	37.6	9075	50.1	
	Gas				14.3
Rankine-Steam	Coal	38.0	8980	38.0	
	Gas				14.3
Field-Steam	Coal	39.8	8580	36.3	
	Gas				13.7

Over a thirty year period at an eight percent interest rate with a fixed percentage charge of fifteen percent for 1000 MW, the additional capital that can be spent for a Field cycle plant is $9 per kilowatt.

V.B. Capital Costs

Capital costs for the three cycles are estimated from data in reference 7, the United Aircraft Research Laboratories report. In the report steam and gas turbine plants of the 1970's and 1980's were evaluated for efficiency and cost. All costs, direct and indirect were considered. The steam and gas turbine plant costs are $165 and $93 per kilowatt respectively.

The cost of the Field plant is based on comparing Field cycle equipment with similar equipment in the other plants. Figure 27 is one possible layout of a 1000 MW Field cycle plant. It has the following characteristics:

Base pressure	200 psia
Maximum pressure	800 psia
Maximum temperature	1200 ^{o}F
Station efficiency	39.8 % (90% heater efficiency)
Net work	122.5 BTU/lbm
Heat input	277.0 BTU/lbm
Flow rate	7.74 x 10^3 lbm/sec
Gaseous turbine output	537 MW
Steam turbine output	464 MW
Steam turbine flow rate	1.50 x 10^3 lbm/sec

Estimates of the sizes of the turbines and compressors indicate that the component sizes are very similar to existing power equipment. Table 4 lists the dimensions of the turbines and compressor. The regenerative gas turbine from reference 7 is also listed. The compressor, operating on wet steam instead of air may require better materials, but the turbine will be operating at lower temperatures, 1200^{o}F vice 2400^{o}F. The mass flow rate through the turbine is high, but it is also at a higher pressure and a lower expansion ratio. For the cost estimate the gaseous turbine and compressor cost is taken as $35 per kilowatt comparable to the 200 and 250 MW

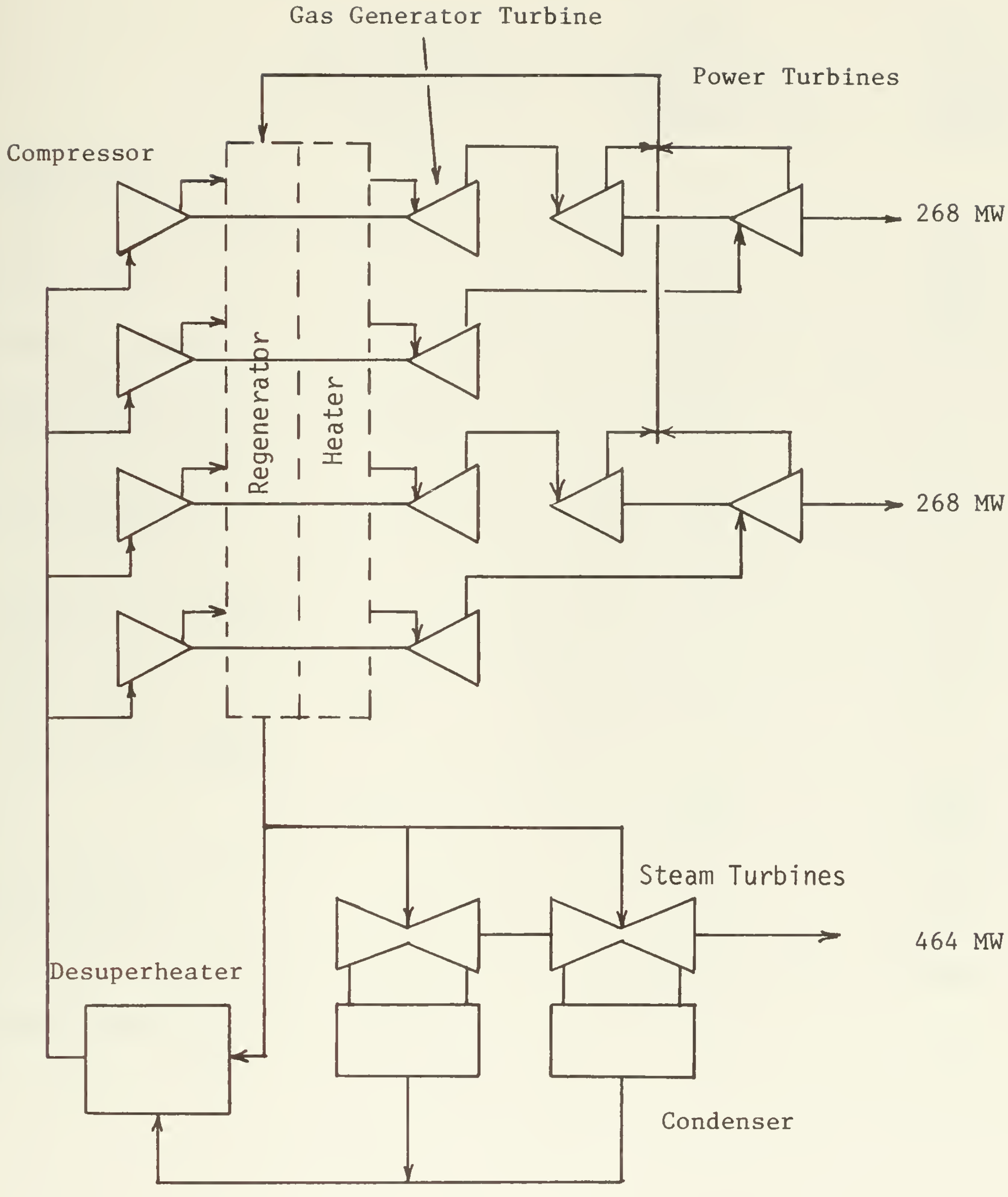

Figure 27. Field Cycle 1000 MW Utility Plant

TABLE 4. SIZES OF FIELD CYCLE AND GAS TURBINE POWER PLANT COMPONENTS

	Gas Turbine 5 200 MW Regenerative Air	Field 4-134 MW Gaseous-Turbine Steam	Field 2-232 MW 2-Flow Steam
Turbine Inlet Temperature, ^{o}F	2400	1200	559
Pressure Ratio	12:1	4:1	200:1
RPM	--	4100	1800
Compressor Stages	14	18	--
Inlet Diameter, ft.	8.9	4.1	--
Blade Height, in.	20	6.5	--
Exit Diameter, ft.	7.5	3.4	--
Blade Height, in.	3.6	2.2	--
Gas Generator Turbine Stages	2	3	--
Outlet Blade Ht., in.	13.7	14.0	--
Power Turbine Stages	2	3	5
RPM	1800	1800	1800
Exit Diameter, Ft.	13	8.46	19.8
Blade Height, in.	24	17.4	53.4
Centrifugal Stress, psi	40,000	30,000	40,000
Exhaust Temperature ^{o}F	866	822	101

gas turbines of reference 7. The cost impact of the gaseous turbine is reduced because it contributes only a part of the work of the cycle (53.7%).

The generator costs for the gaseous turbine section of the plant are assumed to be the same as for the gas turbine plant.

The steam turbine operates at lower pressures and temperatures than the turbines of the steam plant. This reduces the cost of the materials but increases the size of the unit. A Westinghouse price list for steam turbines has a 500 MW turbine cost of about $26 per kilowatt for a 1250 psia turbine [9]. Reference 7 has a cost of about $30 per kilowatt for a 3500 psia turbine. Using the $26 per kilowatt figure and assuming the cost per kilowatt does not vary greatly through the turbine, the cost for the steam turbine for a 1000 MW Field cycle plant is $12.1 per kilowatt.

The regenerator cost is estimated in Appendix B as $18 per kilowatt. The heater cost is estimated as $144 per kilowatt. This does not include the furnace and fuel burning equipment.

The coal burning Field cycle plant land area should be similar to a steam plant as it includes coal storage areas and stacks and the same type of power generation equipment such as turbines, pumps and condensers. The same type boiler is not used, but a furnace is still required as part of the heater.

Neglecting research and development costs the Field cycle design costs are assumed to be the same as the steam plant.

The Field cycle utility plant cost is the sum of the land and component costs as summarized in Table 5 with the steam and gas-turbine plants. One major cost not considered is that of the furnace. It should be less than

TABLE 5. STEAM, GAS TURBINE AND FIELD CYCLE UTILITY PLANT COSTS

Basis: 1980's Technology
Indoor Construction
Escalation 6%
Interest 8%
1000 MW
70% Capacity
1970 Dollars

	Steam	Gas Turbine	Field
Characteristics	3500 psia 1000^oF reheat	2400^oF Inlet 12:1 Regenerative	800/200 psi 1200^oF
Construction Time, years	4	2	4
Efficiency, %	38.0	37.6	39.8
Land Structures	$9.03/KW	4.74	9.03
Station Equipment	12.03	6.82	12.03
Boiler/Heater	54.83	--	144.0
Regenerator	--	--	18.0
Prime Mover	--	49.98	20.4
Turbine Generators	34.20	--	12.1
Generators	--	9.93	5.4
	110.09	71.47	220.96
Other Expenses	1.24	1.25	1.25
	111.33	72.72	222.21
Design and Engineering	13.08	8.00	13.08
	124.41	80.72	235.29
Escalation & Interest	41.18	11.97	77.77
Total Cost Per KW	$165.59	$ 92.69	$313.06

the $55 per kilowatt of the steam plant. Taking it as one-half, the cost per kilowatt of the Field cycle is $341 per kilowatt, or twice that of the steam plant and well over three times that of the gas turbine plant. The reason for this high cost is the heater ($144/KW) as the other component costs are comparable to those of the steam or gas turbine plants. Errors in component cost estimates can not be as significant as the heater cost. The heater cost would have to be reduced to $20 to $30 per kilowatt before the plant cost would approach that of the steam plant. The fuel cost savings of the Field plant ($9/KW) are small when compared to the higher capital costs.

There are two changes that could reduce the heater cost. First, the material cost would be reduced if different materials were used. This is unlikely due to the high temperatures involved. Lower temperatures would lower the efficiency. A two percent loss of efficiency results in a heat rate similar to steam plants.

The second change is the use of a smaller heater. Size is a function of heat transfer required, heat transfer coefficients and temperature differences. To lower the quantity of heat transferred, pressure level, pressure ratio and maximum temperature can be reduced, all of which reduce efficiency. The overall heat transfer coefficient is dominated by the combustion gas heat transfer coefficient which is low and not affected by steam pressure and temperature. The gas temperature must be limited to prevent tube damage. The conclusion is that the Field cycle plant capital cost is much higher than that of steam and gas turbine plants of the near future.

VI. FUTURE OF THE FIELD CYCLE

The station efficiency of the Field cycle power plant is only slightly higher than that of steam and gas-turbine power plants. The fuel savings would not seem to be enough to justify building the higher cost Field cycle plant. The cycle must have a larger efficiency advantage over other cycles or have special advantages in specific applications. The maximum temperature is the parameter that most affects the thermal efficiency of the cycle. If materials can be developed to allow the maximum temperature to be increased, the Field cycle shows a definite advantage over other cycles.

VI.A. High Temperature Materials

The maximum temperature in steam plants is at present limited to about a 1050^{o}F steam temperature. This limit is a compromise between efficiency and economy. Higher superheater temperatures require more expensive metals and the expense must be recovered over the years in lower power production costs. Since projected fuel costs and plant availability are uncertain, the tendency has been to move forward slowly, so slowly in fact, that the maximum temperature of about 1050^{o}F has existed for twenty years. Materials for superheaters above 1050^{o}F are several times the price of materials for lower temperatures. The problems with higher temperatures are two-fold. The high temperature increases the external tube corrosion for both coal and fuel-oil, and increases internal corrosion from the steam itself.

External corrosion of superheaters of coal burning plants is due to alkalies and sulfur oxides that are deposited on the tubes during combustion in the fly ash. Chemical reactions occur between the alkalies, sulfur oxides and the fly ash forming complex alkali sulfates. The formation

rate of these complex alkali sulfates is temperature dependent, the maximum occuring at a metal temperature of 1250 to 1350° where the alkali sulfates are liquid and attack the superheater metal. To limit coal-ash corrosion, careful furnace design is necessary to maintain low ash deposition rates and low gas temperatures. Experience indicates that there is a relation between stable and corrosive coal-ash and the gas temperature. Lower gas temperatures permit higher metal temperatures (to about 1200°F). All bituminous coals contain sufficient sulfur and alkali metals for corrosive ash deposit formation, those with more than 3.5 percent sulfur are the worst. Other methods to reduce corrosion are, in addition to improved combustion and better coals, stainless steel shields for problem superheat areas, and the use of corrosion-resistant alloys and ceramic coatings.

Furnaces using fuel oil are also subject to corrosion attack on high temperature superheater tubes. Vanadium oxides are formed during combustion from the vanadium compounds in the fuel oil. Sodium in the fuel reacts with sulfur oxides to form sodium sulfates. The vanadium oxides and sodium sulfates combine to form sodium-vanadium complexes that have lower melting points than either of the original compounds, ranging from 480 to 1250°F. When molten, these complexes are corrosive. Sulfates are also present in fuel oil and can corrode the tubes. As in coal ash corrosion, low gas temperatures are beneficial. If the vanadium content is kept low, oil-ash corrosion is minimized. Methods of reducing fuel-ash corrosion in boilers are careful selection or purification of fuel oils to maintain low fuel ash components in the fuel oil. Proper design and operation of the boiler can be successful in keeping the fuel-ash from being deposited on tube surfaces. Using soot blowers removes the ash. Good combustion with minimum excess air has been shown to reduce vanadium and sulfur oxide formation.

Natural gas fuels have proven to be better than coal or fuel oil with regard to superheater corrosion since the gas contains neither vanadium or sulfur.

To summarize, external corrosion is due to various chemical complexes deposited on the tubes during operation. These complexes are molten in the range of metal temperatures of 1200 to 1350^{o}F and attack the tube material. The corrosion rate is sufficiently high to cause rapid failure of the tubes in relatively short periods of time. Those materials that are most resistant to attack are the 18-8 stainless steels, the present superheater materials. The effort to reduce corrosion is directed toward better design and operation vice better materials. [10]

The effect of high temperature steam on the internal surfaces of the superheater tubes has been investigated by the American Society of Mechanical Engineers with the support of the power industry and government [11]. Test sections of various superheater materials were exposed to steam temperatures of 1100, 1200, 1350, and 1500^{o}F for periods up to three years. The properties of the superheater material and the effect of heat transfer were determined. The test results indicated that scale thickness increased rapidly for steam temperatures over 1150^{o}F. A high chromium content for the chromium molybdenum steels was observed to help the corrosion resistance of the steel. Figure 28 from reference 11 shows the effects on various Cr-Mo steels. The least affected is the 9Cr-1Mo steel. The ferritic chromium-molybdenum steel had generally uniform scale thicknesses that remained adhered to the inner tube surface. The austenitic tubes had varied scale thicknesses and penetrations, so much so that definite evaluation of scaling rate growth was not possible. General trends seemed to indicate that the high chromium steels such as 310 (25Cr-20Ni) and Incoloy and Inconel had lower scaling rates. Also, it was noted that mach-

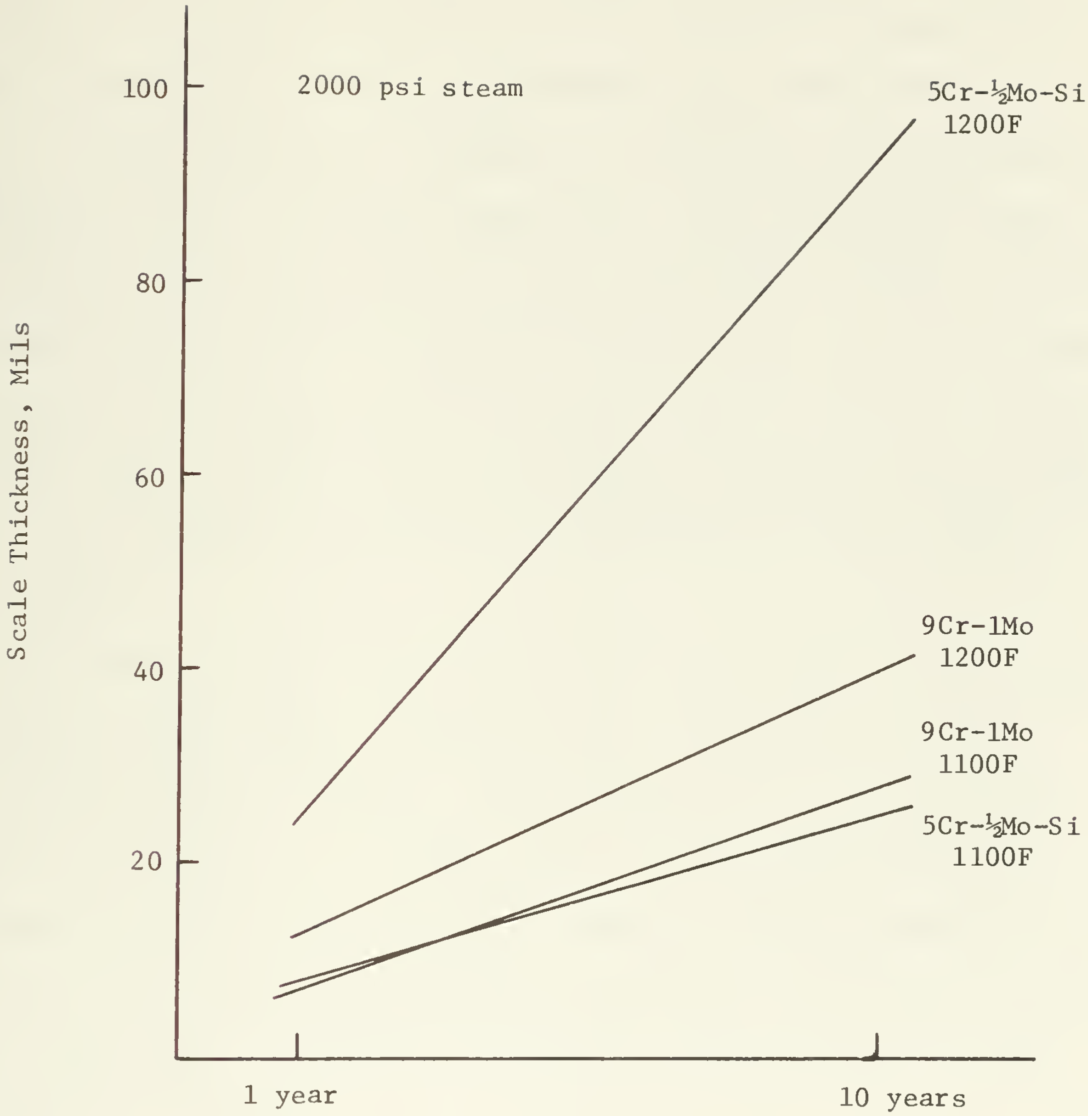

Figure 28. Scale Thicknesses of Ferritic Steels in High Temperature Steam [11].

ining the surface of type 304 increased the resistance to corrosion. Pressure has little effect on corrosion rate. The maximum steam temperature for any steel considered is on the order of 1150°F for external corrosion for fuel oil and 1300°F for coal and 1200°F for internal corrosion. These are all present day applications and depend greatly on design and gas temperatures. The maximum of 1050°F for operating plants is apparently based on the above numbers with a small margin for error.

Future trends for the industry are not encouraging. Little advance is expected for many years in power plant applications. The 1970 National Power Survey by the Federal Power Commission points out the cost of going to 1200°F does not justify the slight increase in efficiency and that minor pressure and temperature increases may be expected, but major increases are not [1].

The industry oriented view is that it is possible that various types of alloys may permit higher temperatures, but that anything developed will be prohibitively expensive. Since so much effort has already been placed on steam plants it is unlikely that any cheap material has been overlooked. The advantage of the Field cycle in regard to temperature is that the lower pressures involved place less stress on the tube, and thus, should permit higher temperatures. However, external corrosion exists irregardless of pressure.

Experiences in Europe with closed-cycle gas-turbine plants (CCGT) using air as the fluid indicates that high temperatures are possible with careful design and fuel treatment [12]. The CCGT plants are smaller, 15-20 MW, than steam utility plants in the United States but they are operating at air temperatures of up to 1300°F. The airheaters of these

CCGT plants are usually vertical, two-pass, top-fired units. The fuel has been either coal or oil. The tubes are austenitic materials of Cr-Ni and Nb, Mo or Co. A 14.3 MW plant in Germany has tubes of 19Cr 13Ni 10Co 2Mo and 3Nb with other sections of 16Cr 16Ni 2Mo and 1Nb. The fuels for these plants are treated to reduce the external corrosion. Tests conducted by Escher Wyss used dolomite powder $CaMg(CO_3)_2$ suspended in light oil added to heavy commercial fuel oil [12]. The test lasted for 3200 hours and resulted in no evidence of low or high temperature external corrosion. The air temperatures were over 1200°F.

These tests and CCGT experience demonstrate that higher external temperatures are possible with careful design, fuel treatment and proper materials. If this experience can be transferred to plants that use steam instead of air is not known but advances are being made in high temperature tube-type heaters similar to the type the Field cycle requires.

VI.B. Nuclear Applications

The Field cycle may have specific applications to nuclear power, much more so than trying to replace present steam and gas-turbine power plants. The high thermal efficiencies and lower pressures of the Field cycle contribute to solving problems in steam-cooled-breeder reactors, namely, fuel cladding integrity during normal and emergency conditions. The reduced pressure decreases the strength requirements of the fuel cladding during fuel burnup. A loss of coolant accident with low density steam is believed to be less severe than for higher pressure systems. In reference 13 these considerations are discussed fully and the use of the Field cycle in steam-cooled-breeder reactors is suggested as a possible solution to breeder reactor problems.

CHAPTER VII. CONCLUSIONS AND RECOMMENDATIONS

The gaseous-steam cycle or Field cycle has been evaluated to determine the effects of variations in its major parameters on thermal efficiency. The maximum temperature was found to have the most significant influence on efficiency. Base pressure levels of 200 psia and pressure ratios of 4:1 or greater are the upper limit if the moisture content of the steam turbine is to be kept low. A process steam heating system instead of the steam turbine would permit higher pressures and pressure ratios to be used. Reheat gaseous turbines would add one to two percentage points of efficiency, but would decrease the moisture content of the exhaust steam very little.

The maximum temperature depends on materials and steam utility plants in the United States have been operating with temperatures $1050^{o}F$ for some time. European experience with closed cycle gas turbine plants using air at temperatures of $1300^{o}F$ may be applicable to the design and operation of heaters for high temperature Field cycle plants.

A preliminary cost estimate of a Field cycle power plant resulted in a high capital cost with the major factor the heater cost due to the materials. The high capital cost is not offset by lower fuel costs resulting from a higher thermal efficiency.

The next step in the evaluation of the gaseous-steam or Field cycle is a detailed design of the components and a determination of their cost. Compact heat exchanger design methods should reduce the size and cost of the regenerator and heater and make the Field cycle capital cost more comparable to other power plants. Other applications should be explored such as transportation and process steam heating.

Small scale experimental Field cycle plants need to be designed, built and operated to gain experience in component performance and control.

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APPENDIX A. FIELD CYCLE COMPUTER PROGRAM

The computer program written to evaluate the Field cycle uses basic thermodynamic definitions to calculate the properties at each point in the cycle. Table A-2 lists the points of the cycle in the order in which they appear in the program, the input and output variables, and the specific formulas used. With the required properties at each point, the work and heat values were determined with the formulas as listed in Table A-3. The basic program for a given pressure level evaluated for changing pressure ratios is included at the end of this Appendix. This program is easily modified for the different parameters.

The basic main program is uncomplicated, but requires the determination of the thermodynamic properties at each cycle point. Subroutines for water and steam properties are available, and the one found most suited was subroutine WASP [14]. Other subroutines were not used due to lack of coverage of subcooled liquid areas or due to their complexity.

The subroutine WASP (Water And Steam Properties) was developed by Hendricks, Peller and Baron of the Lewis Research Center. The subroutine will calculate temperature, pressure, density, enthalpy, entropy, specific heat (C_p and C_v), sonic velocity, $(\partial P/\partial \rho)_T$, $(\partial P/\partial T)_\rho$, viscosity, thermal conductivity, surface tension, and the Laplace constant. The subroutine uses as input any two of temperature, pressure and density or pressure and enthalpy or pressure and entropy. The Lewis Research Center developed the subroutine to cover a wide range and to be adaptable for a variety of uses. The temperature range is from the triple point to $1750^{o}K$, the pressure range is from .1 to 100 MN/M^2 (1 to 1000 bars). Subroutine WASP contains basic thermodynamic equations from Keyes, Keenan, Hill and Moore [15], Schmidt [16], and the ASME Steam Tables [17] and others developed by the authors to account for areas near the critical point, and to ensure

the subroutine provides a continuous set of values. The basic equation is $\psi = \psi$ (p, t) where ψ is the Helmholtz free energy. This equation is expanded and used for the thermodynamic properties. A complete listing and discussion of the equations in the subroutine is contained in the subroutine WASP report.

For this study the subroutine was modified by eliminating those subroutine sections that evaluated properties not needed in the cycle calculations. Certain input variations were not used so they were also removed. Those sections remaining were changed as little as possible so that the additional sections not included could be added if desired to evaluate other properties. It should be noted that below a pressure of 1 bar (14.5 psia), pressure cannot be used as an input, but temperature can, and the subroutine will return the correct pressure if working in the saturated region. The subroutine WASP report contains a users guide for the program.

The main program and subroutine WASP were run on an IBM 370. Changes that were made were the:

1. Use of single precision over double precision to reduce cost.

Two test statements were changed in subroutine TEMPPH from 10^{-5} to 4×10^{-4} to permit the correct choice to be made. Other such problems may exist in the unused sections.

2. Addition of variable KR to the argument of subroutine ENTH and ENT to allow the variable to be passed through the subroutines.

Subroutine WASP proved to be flexible and accurate. It will calculate values in all regions, liquid, saturated and superheated and contains internal checks for out of range inputs and outputs.

Table A - 1. Computer Program Variables

Variable	Name
D	Density, lbm/ft^3
DELP1	Pressure loss in high pressure side of regenerator and heater, psia
DELP2	Pressure loss in low pressure side of regenerator, psia
DL	Density, saturated liquid, lbm/ft^3
DV	Density, saturated vapor, lbm/ft^3
EFFC	Compressor isentropic efficiency
EFFECT	Regenerator effectiveness
EFFGT	Gaseous turbine isentropic efficiency
EFFP	Pump isentropic efficiency
EFFST	Steam turbine isentropic efficiency
FHFR	Feedheater fraction (of temperature difference between base pressure saturation temperature and condensing temperature)
FIELDE	Field cycle thermal efficiency
H	Enthalpy, BTU/lbm
HA	Enthalpy at point in cycle, BTU/lbm
HL	Enthalpy, saturated liquid, BTU/lbm
HR	Enthalpy, saturated vapor at condensing temperature, BTU/lbm
HR1	Enthalpy, saturated vapor, of feedheater, BTU/lbm
HV	Enthalpy, saturated vapor, BTU/lbm
KR	Code for liquid, saturated or superheated region in subroutine WASP
KU	Code for system of units in subroutine WASP
MDOT	Mass flow fraction through steam turbine
MDOT1	Mass flow fraction through condenser

Table A - 1. Computer Program Variables - Continued

Variable	Name
MDOT2	Mass flow fraction of steam to feedheater
MDOT5	Mass flow fraction of steam to desuperheater
P	Pressure, psia
PCOND	Condenser pressure, psia
PHIGH	Maximum pressure, psia
PLOW	Base pressure level, psia
PR	Pressure ratio
QIN	Heat transfer to heater, BTU/lbm
QREG	Heat transfer in regenerator, BTU/lbm
QREJ	Heat transfer from condenser, BTU/lbm
QUAL	Quality of steam turbine exhaust. Also used when determining, compressor inlet conditions.
QUAL1	Quality of bleed stream to feedheater
S	Entropy, BTU/lbm- ^{o}R
SA	Entropy at point in cycle, BTU/lbm- ^{o}R
SL	Entropy, saturated liquid, BTU/lbm- ^{o}R
SR	Entropy, saturated vapor at condensing temperature, BTU/lbm- ^{o}R
SR1	Entropy, saturated vapor, of feedheater, BTU/lbm- ^{o}R
SV	Entropy, saturated vapor, BTU/lbm- ^{o}R
T	Temperature, ^{o}F
TMAX	Maximum temperature, ^{o}F
V	Specific volume, ft^3/lbm
VR	Specific volume, saturated vapor at condensing temperature, ft^3/lbm
VR1	Specific volume, saturated vapor, of feedheater, ft^3/lbm

Table A - 1. Computer Program Variables - Continued

Variable	Name
WC	Compressor work, BTU/lbm
WGT	Gaseous turbine work, BTU/lbm
WNET	Net work of cycle, BTU/lbm
WP	Net pump work, BTU/lbm (of total flow rate)
WP1	Pump 1 work, BTU/lbm (of total flow rate)
WP2	Pump 2 work, BTU/lbm (of total flow rate)
WST	Steam turbine work, BTU/lbm (of total flow rate)

Subscripts

Subscripts for temperature, pressure, enthalpy, entropy and specific volume correspond to the cycle points as depicted in Figure 17.

Table A-2. Main Program Calculations

Variables as defined in Table A-1. Cycle points from Figure 17.

Cycle Point	Position	Input	Output	Remarks
1	Condenser outlet, Pump inlet	T(1)	P(1), HA(1) SA(1), V(1) HR, SR, VR	No Subcooling
6	Compressor outlet, isentropic	P(6)	T(6), HA(6) SA(6), V(6)	Saturated vapor P(6) = PHIGH
5	Compressor inlet	PLOW, SA(6)	T(5), HA(5) V(5)	
7	Compressor outlet, actual	HA(5), HA(6) EFFC	HA(7)	$\frac{HA(6)-HA(5)}{HA(7)-HA(5)}$
9	Heater outlet, Gaseous turbine inlet	PHIGH, TMAX	HA(9), SA(9) V(9)	PHIGH reduced by pressure loss DELP1
10	Gaseous turbine outlet, isentropic	PLOW, SA(9)	HA(10) isentropic	PLOW increased by pressure loss DELP2
10	Gaseous turbine outlet, actual	HA(10) isentropic, HA(9), EFFGT	HA(10)	$\frac{HA(9)-HA(10)\ \text{isentropic}}{HA(9)-HA(10)}$
10	Gaseous turbine outlet, actual	PLOW, HA(10)	T(10), SA(10) V(10)	PLOW increased by pressure loss DELP2
11	Regenerator outlet, Steam turbine inlet	T(10), T(7) EFFECT	T(11)	$\frac{T(10)-T(11)}{T(10)-T(7)}$
11	Regenerator outlet, Steam turbine inlet	PLOW, T(11)	HA(11), SA(11) V(11)	

Table A-2. Continued

Cycle Point	Position	Input	Output	Remarks
13	Steam turbine outlet, isentropic	T(1), P(1) SA(11)	HA(13) isentropic QUAL	Test for saturated or superheated region
13	Steam turbine outlet, actual	HA(13), isentropic, HA(11), EFFST	HA(13), SA(13) V(13), QUAL	$\frac{HA(11)-HA(13)\ \text{isentropic}}{HA(11)-HA(13)}$
8	Regenerator outlet, Heater inlet	HA(7), HA(10), HA(11)	HA(8)	HA(8)-HA(7)=HA(10)-HA(11)
8	Regenerator outlet, Heater inlet	P(7), P(9), HA(7), HA(8), HA(9)	P(8)	$\frac{P(8)-P(7)}{P(9)-P(7)} = \frac{HA(8)-HA(7)}{HA(9)-HA(7)}$
8	Regenerator outlet, Heater inlet	P(8), HA(8)	T(8), SA(8), V(8)	
3	Feedheater	T(5), T(13), FHFR	T(3)	T(3) = FHFR*(T(5)-T(13)) + T(13)
3	Feedheater	T(3)	P(3), HA(3), SA(3), V(3), HR1, SR1, VR1	Saturated liquid
2	Pump outlet, Feedheater inlet	P(2), SA(1)	HA(2) isentropic	P(2) = P(3)
2	Pump outlet, Feedheater inlet	HA(2) isentropic, HA(1), EFFP	HA(2)	$\frac{HA(2)-HA(1)\ \text{isentropic}}{HA(2)-HA(1)}$
2	Pump outlet, Feedheater inlet	P(2), HA(2)	T(2), SA(2), V(2)	
4	Pump #2 outlet, Desuperheater inlet	PLOW, SA(3)	HA(4) isentropic	

Table A-2. Continued

Cycle Point	Position	Input	Output	Remarks
4	Pump #2 outlet, Desuperheater inlet	HA(4) isentropic, HA(3), EFFP		$\frac{\text{HA(4)isentropic}-\text{HA(3)}}{\text{HA(4)}-\text{HA(3)}}$
4	Pump #2 outlet, Desuperheater inlet	PLOW, HA(4)	T(4), SA(4), V(4)	P(4) = PLOW
12	Bleed steam	P(3), SA(11)	HA(12) isentropic QUAL1	Test for saturated or superheated region
12	Bleed steam	HA(12) isentropic, HA(11), EFFST	HA(12), SA(12) V(12), QUAL1	$\frac{\text{HA(11)}-\text{HA(12)}}{\text{HA(11)}-\text{HA(12)}}$ isentropic $\text{QUAL1} = \frac{\text{HA(12)}-\text{HA(3)}}{\text{HR1}-\text{HA(3)}}$

Table A-3. Heat and Work

Name	Variable	Formula
Steam Turbine Flow Fraction	MDOT	$\frac{HA(11)-HA(5)}{HA(11)-HA(4)}$
Condenser Flow Fraction	MDOT1	$\frac{HA(12)-HA(3)}{HA(12)-HA(2)}$
Bleed Steam Flow Fraction	MDOT2	MDOT-MDOT1
Desuperheater Steam Flow Fraction	MDOT5	1.0 - MDOT
Gaseous Turbine Work	WGT	HA(9)-HA(10)
Steam Turbine Work	WST	MDOT* (HA(11)-HA(12)) + MDOT1* (HA(12)-HA(13))
Compressor Work	WC	HA(7) - HA(5)
Pump Work	WP1	HA(2) - HA(1)
Pump Work	WP2	HA(4) - HA(3)
Total Pump Work	WP	WP1 + WP2
Heat Input	QIN	HA(9) - HA 8)
Heat Rejected	REJ	MDOT1* (HA(13)-HA(1))
Heat Regenerated	QREG	HA(8)-HA(7) = HA(10)-HA(11)
Net Work	WNET	WGT = WST - WC - WP
Thermal Efficiency	FIELDE	$\frac{WNET}{QIN}$

TABLE A-4. DATA - BASE PRESSURE LEVEL 100 PSIA
PRESSURE RATIO 2
MAXIMUM TEMPERATURE 1200 F
PRESSURE LOSS 5%
ONE FEEDHEATER
CYCLE POINT NUMBERS FROM FIGURE 17

	P-PSIA	T-F	H-BTU/LBM	V-FT3/LBM	S-BTU/LBM-R
1	1.00	101.74	69.78	0.01614	0.1327
2	15.54	101.75	69.84	0.01614	0.1327
3	15.54	214.80	182.98	0.01673	0.3163
4	100.00	214.94	183.31	0.01673	0.3164
5	100.00	327.86	1142.95	4.21121	0.5464
6	200.00	381.85	1199.30	2.28922	1.5464
7	200.00	391.59	1205.56	2.32831	1.5539
8	196.73	919.62	1487.39	4.13221	1.8149
9	195.00	1200.00	1635.84	5.04451	1.9137
10	102.50	1010.00	1537.24	8.50517	1.9212
11	100.00	453.43	1255.41	5.28686	1.6832
12	15.54	214.80	1127.53	24.81215	1.7166
13	1.00	101.74	987.58	295.22070	1.7673

WGT =	98.60	BTU/LBM	MDOT	=	0.1049
WST =	26.28	BTU/LBM	MDOT1	=	0.0920
WC =	62.61	BTU/LBM	MDOT2	=	0.0129
WP =	0.04	BTU/LBM	MDOT5	=	0.8951
WNET =	62.23	BTU/LBM	QUAL1	=	0.9752
QIN =	148.44	BTU/LBM	QUAL	=	0.8859
QREJ =	84.40	BTU/LBM	FIELD2	=	0.4192
QREG =	281.83	BTU/LBM			

TABLE A-4. DATA - BASE PRESSURE LEVEL 100 PSIA
PRESSURE RATIO 3
MAXIMUM TEMPERATURE 1200 F
PRESSURE LOSS 5%
ONE FEEDHEATER
CYCLE POINT NUMBERS FROM FIGURE 17

	P-PSIA	T-F	H-BTU/LBM	V-FT3/LBM	S-BTU/LBM-R
1	1.00	101.74	69.78	0.01614	0.1327
2	15.54	101.75	69.84	0.01614	0.1327
3	15.54	214.80	182.98	0.01673	0.3163
4	100.00	214.94	183.31	0.01673	0.3164
5	100.00	327.86	1115.45	4.07463	1.5115
6	300.00	417.40	1203.87	1.54406	1.5115
7	300.00	431.37	1213.70	1.58460	1.5227
8	296.21	808.82	1425.82	2.49335	1.7239
9	292.50	1200.00	1633.92	3.35452	1.8682
10	102.50	899.55	1480.18	7.85353	1.8809
11	100.00	478.19	1268.06	5.44687	1.6968
12	15.54	214.80	1137.26	25.06779	1.7310
13	1.00	101.74	996.00	297.93040	1.7823

WGT =	153.74	BTU/LBM	MDOT =	0.1407	
WST =	35.85	BTU/LBM	MDOT1 =	0.1235	
WC =	98.24	BTU/LBM	MDOT2 =	0.0172	
WP =	0.05	BTU/LBM	MDOT5 =	0.8593	
WNET =	91.29	BTU/LBM	QUAL1 =	0.9852	
QIN =	208.10	BTU/LBM	QUAL =	0.8940	
QREJ =	114.39	BTU/LBM	FIELDE =	0.4387	
QREG =	212.12	BTU/LBM			

TABLE A-4. DATA - BASE PRESSURE LEVEL 100 PSIA
PRESSURE RATIO 4
MAXIMUM TEMPERATURE 1200 F
PRESSURE LOSS 5%
ONE FEEDHEATER
CYCLE POINT NUMBERS FROM FIGURE 17

	P-PSIA	T-F	H-BTU/LBM	V-FT3/LBM	S-BTU/LBM-R
1	1.00	101.74	69.78	0.01614	0.1327
2	15.54	101.75	69.84	0.01614	0.1327
3	15.54	214.80	182.98	0.01673	0.3163
4	100.00	214.94	183.31	0.01673	0.3164
5	100.00	327.86	1095.00	3.97303	1.4856
6	400.00	444.67	1205.45	1.16186	1.4856
7	400.00	460.77	1217.72	1.19961	1.4990
8	396.03	735.92	1382.32	1.72877	1.6575
9	390.00	1200.00	1631.99	2.50951	1.8356
10	102.50	825.30	1442.32	7.41366	1.8522
11	100.00	497.22	1277.72	5.56882	1.7070
12	15.54	214.80	1144.56	25.25926	1.7419
13	1.00	101.74	1002.32	299.96140	1.7936

WGI =	189.68	BTU/LBM	MDOT	=	0.1670
WST =	43.10	BTU/LB4	MDOT1	=	0.1467
WC =	122.73	BTU/LBM	MDOT2	=	0.0203
WP =	0.06	BTU/LBM	MDOT5	=	0.8330
WNET =	109.99	BTU/LBM	QUAL1	=	0.9928
QIN =	249.67	BTU/LBM	QUAL	=	0.9001
QREJ =	136.80	BTU/LBM	FIELDE	=	0.4405
QREG =	164.60	BTU/LBM			

TABLE A-4. DATA - BASE PRESSURE LEVEL 100 PSIA
PRESSURE RATIO 5
MAXIMUM TEMPERATURE 1200 F
PRESSURE LOSS 5%
ONE FEEDHEATER
CYCLE POINT NUMBERS FROM FIGURE 17

	P-PSIA	T-F	H-BTU/LBM	V-FT3/LBM	S-BTU/LBM-R
1	1.00	101.74	69.78	0.01614	0.1327
2	15.54	101.75	69.84	0.01614	0.1327
3	15.54	214.80	182.98	0.01673	0.3163
4	100.00	214.94	183.31	0.01673	0.3164
5	100.00	327.86	1078.38	3.89049	1.4645
6	500.00	467.09	1205.31	0.92821	1.4645
7	500.00	484.27	1219.42	0.96273	1.4796
8	496.08	684.44	1348.12	1.29303	1.6046
9	487.50	1200.00	1630.06	2.00249	1.8101
10	102.50	769.95	1414.31	7.08442	1.8299
11	100.00	512.84	1285.60	5.66828	1.7152
12	15.54	214.80	1150.43	25.41338	1.7506
13	1.00	101.74	1007.40	301.59660	1.8026

WGT =	215.75	BTU/LBM	MDOT	=	0.1880
WST =	49.06	BTU/LBM	MDOT1	=	0.1653
WC =	141.04	BTU/LBM	MDOT2	=	0.0227
WP =	0.07	BTU/LBM	MDOT5	=	0.8120
WNET =	123.69	BTU/LBM	QUAL1	=	0.9988
QIN =	281.93	BTU/LBM	QUAL	=	0.9050
QREJ =	155.00	BTU/LBM	FIELDE	=	0.4387
QREG =	128.70	BTU/LBM			

TABLE A-4. DATA - BASE PRESSURE LEVEL 100 PSIA
PRESSURE RATIO 6
MAXIMUM TEMPERATURE 1200 F
PRESSURE LOSS 5%
ONE FEEDHEATER
CYCLE POINT NUMBERS FROM FIGURE 17

	P-PSIA	T-F	H-BTU/LBM	V-FT3/LBM	S-BTU/LBM-R
1	1.00	101.74	69.78	0.01614	0.1327
2	15.54	101.75	69.84	0.01614	0.1327
3	15.54	214.80	182.98	0.01673	0.3163
4	100.00	214.94	183.31	0.01673	0.3164
5	100.00	327.86	1064.17	3.81993	1.4464
6	600.00	486.30	1204.05	0.77010	1.4464
7	600.00	503.93	1219.59	0.80164	1.4627
8	596.33	647.30	1319.53	1.01441	1.5604
9	585.00	1200.00	1628.10	1.66447	1.7891
10	102.50	726.11	1392.24	6.82264	1.8117
11	100.00	526.15	1292.31	5.75268	1.7220
12	15.54	222.61	1155.36	25.76350	1.7579
13	1.00	101.74	1011.67	302.97090	1.8102

WGT =	235.86	BTU/LBM	MDOT	=	0.2057
WST =	54.18	BTU/LBM	MDOT1	=	0.1810
WC =	155.42	BTU/LBM	MDOT2	=	0.0247
WP =	0.08	BTU/LBM	MDOT5	=	0.7943
WNET =	134.54	BTU/LBM	QUAL1	=	1.0000
QIN =	308.58	BTU/LBM	QUAL	=	0.9092
QREJ =	170.48	BTU/LBM	FIELDE	=	0.4360
QREG =	99.93	BTU/LBM			

TABLE A-4. DATA - BASE PRESSURE LEVEL 100 PSIA
PRESSURE RATIO 7
MAXIMUM TEMPERATURE 1200 F
PRESSURE LOSS 5%
ONE FEEDHEATER
CYCLE POINT NUMBERS FROM FIGURE 17

	P-PSIA	T-F	H-BTU/LBM	V-FT3/LBM	S-BTU/LBM-R
1	1.00	101.74	69.78	0.01614	0.1327
2	15.54	101.75	69.84	0.01614	0.1327
3	15.54	214.80	182.98	0.01673	0.3163
4	100.00	214.94	183.31	0.01673	0.3164
5	100.00	327.86	1051.62	3.75758	1.4305
6	700.00	503.20	1201.98	0.65585	1.4305
7	700.00	520.91	1218.68	0.68468	1.4477
8	696.74	620.69	1294.62	0.82241	1.5221
9	682.50	1200.00	1626.14	1.42301	1.7712
10	102.50	689.96	1374.11	6.60602	1.7961
11	100.00	537.81	1298.17	5.82640	1.7280
12	15.54	231.42	1159.63	26.12025	1.7641
13	1.00	101.74	1015.37	304.16110	1.8168

WGT =	252.03	BTU/LBM	MDOT	=	0.2212
WST =	58.72	BTU/LBM	MDOT1	=	0.1947
WC =	167.07	BTU/LBM	MDOT2	=	0.0265
WP =	0.08	BTU/LBM	MDOT5	=	0.7789
WNET =	143.60	BTU/LBM	QUAL1	=	1.0000
QIN =	331.52	BTU/LBM	QUAL	=	0.9127
QREJ =	184.10	BTU/LBM	FIELDE	=	0.4332
QREG =	75.94	BTU/LBM			

```
C      FIELD CYCLE - COMBINATION OF BRAYTON AND RANKINE CYCLES.
C      INCLUDES COMPONENT EFFICIENCIES, PRESSURE LOSSES AND ONE
C      FEEDHEATER.
C      SUBSCRIPTS CORRESPOND TO FIGURE 17.
C      UTILIZES SUBROUTINE WASP TO CALCULATE PROPERTIES.
C      THIS PROGRAM WILL CALCULATE THERMAL EFFICIENCY FOR THE GIVEN BASE
C      PRESSURE LEVEL AND PRESSURE RATIOS AS LISTED IN DATA.
C
       DIMENSION T(16), P(16), V(16),SA(16),HA(16)
       COMMON/PROPTY/KU,DL,DV,HL,HV,S,SL,SV
       REAL MDOT,MDOT1,MDOT2,MDOT5
C
C      SPECIFY UNITS
       KU = 3
C
C      COMPONENT EFFICIENCIES
       EFFGT = .9
       EFFST = .85
       EFFC = .9
       EFFP = .8
       EFFECT = .9
C
C      BASE PRESSURE LEVEL - PSIA
       PLOW = 100.
C
C      MAXIMUM TEMPERATURE - DEGREES F
       TMAX = 1659.67
       DO 10 I = 1,16
    10 T(I) = 0.0
C
C      CONDENSING TEMPERATURE - DEGREES F
       T(1) = 101.74 + 459.67
C
       PCOND = 0.0
       KR = 1
       CALL WASP (1,3,T(1),PCOND,D,H,KR)
```

```
      T(1) = T(1)-459.67
      P(1) = PCOND
      V(1) = 1./DL
      SA(1) = SL
      HA(1) = HL
      HR = HV
      SR = SV
      VR = 1./DV
C
C       PRESSURE RATIO
C     PRESSURE RATIOS ARE LISTED ON DATA CARDS
    5 READ,PR
      IF(PR.EQ.0.)GO TO 70
      PHIGH = PR*PLOW
C
C     PRESSURE LOSSES
      DELP1 = .025*PHIGH
      DELP2 = .025*PLOW
C
      P(6) = PHIGH
      KR = 1
      CALL WASP (1,3,T(6),P(6),D,H,KR)
      T(6) = T(6) -459.67
      V(6) = 1./DV
      SA(6) = SV
      HA(6) = HV
C
      P(5) = PLOW
      KR = 0
      S = SA(6)
      CALL WASP (5,3,T(5),P(5),D,H,KR)
      QUAL = (S-SL)/(SV-SL)
      H = HL+ QUAL*(HV-HL)
      HA(5) = H
      T(5) = T(5)-459.67
      D = (DV*DL)/(DV+QUAL*(DL-DV))
```

```
      V(5) = 1./D
      SA(5) = S
C
      HA(7) = HA(5)+(HA(6)-HA(5))/EFFC
      P(7) = PHIGH
      KR = 0
      CALL WASP (4,2,T(7),P(7),D,HA(7),KR)
      T(7) = T(7)-459.67
      V(7) = 1./D
      SA(7) = S
C
      P(9) = PHIGH-DELP1
      T(9) = TMAX
      KR = 0
      CALL WASP (1,3,T(9),P(9),D,HA(9),KR)
      T(9) = T(9)-459.67
      V(9) = 1./D
      SA(9) = S
C
      P(10) = PLOW+DELP2
      KR = 0
      CALL WASP (5,1,T(10),P(10),D,HA(10),KR)
      HA(10) = EFFGT*(HA(10)-HA(9))+HA(9)
      KR = 0
      CALL WASP (4,2,T(10),P(10),D,HA(10),KR)
      T(10) = T(10)-459.67
      V(10) = 1./D
      SA(10) = S
C
      T(11) = T(10)-EFFECT*(T(10)-T(7))
      IF (T(10).GE.T(7)) GO TO 8
      WRITE(6,9)
    9 FORMAT ('0',' NO REGENERATION')
      GO TO 7
    8 P(11) = PLOW
      T(11) = T(11)+459.67
```

```
      KR = 0
      CALL WASP (1,3,T(11),P(11),D,H,KR)
      T(11) = T(11)-459.67
      V(11) = 1./D
      HA(11) = H
      SA(11) = S
      IF(SA(11).GT.SR) GO TO 90
C
      T(13) = T(1)
      P(13) = P(1)
      QUAL = (SA(11)-SA(1))/(SR-SA(1))
      HA(13) = HA(1)+QUAL*(HR-HA(1))
      HA(13) = EFFST*(HA(13)-HA(11))+HA(11)
      IF (HA(13).GT.HR)GO TO 90
      QUAL = (HA(13)-HA(1))/(HR-HA(1))
      SA(13) = SA(1)+QUAL*(SR-SA(1))
      V(13) = V(1) +QUAL*(VR-V(1))
C
      HA(8) = HA(10)-HA(11)+HA(7)
      P(8) = P(7)+(P(9)-P(7))*(HA(8)-HA(7))/(HA(9)-HA(7))
      KR = 0
      CALL WASP (4,2,T(8),P(8),D,HA(8),KR)
      V(8) = 1./D
      T(8) = T(8)-459.67
      SA(8) = S
C
C     FEEDHEATER
      FHFR = .5
      IF(T(3).GT.0.0) GO TO 21
      T(3) = (T(5)-T(13))*FHFR+459.67+T(13)
      P(3) = 0.0
      KR = 1
      CALL WASP(1,3,T(3),P(3),D,H,KR)
      HA(3) = HL
      SA(3) = SL
      V(3) = 1./DL
```

```
      T(3) = T(3)-459.67
      HR1 = HV
      SR1 = SV
      VR1 = 1./LV
C
      P(2) = P(3)
      S = SA(1)
      KR = 0
      CALL WASP (5,1,T(2),P(2),D,H,KR)
      HA(2) = H
      HA(2) = HA(1) +(HA(2)-HA(1))/EFFP
      KR = 0
      CALL WASP(4,2,T(2),P(2),D,HA(2),KR)
      T(2) = T(2) -459.67
      V(2) = 1./D
      SA(2) = S
C
      P(4) = PLOW
      S = SA(3)
      KR = 0
      CALL WASP(5,1,T(4),P(4),D,H,KR)
      HA(4) = H
      HA(4) = HA(3)+(HA(4)-HA(3))/EFFP
      KR = 0
      CALL WASP(4,2,T(4),P(4),D,HA(4),KR)
      T(4) = T(4)-459.67
      V(4) = 1./D
      SA(4) = S
C
   21 P(12) = P(3)
      S = SA(11)
      IF (S.GT.SR1) GO TO 30
      QUAL1 = (SA(11)-SA(3))/(SR1-SA(3))
      HA(12) = HA(3)+QUAL1*(HR1-HA(3))
      HA(12) = EFFSI*(HA(12)-HA(11))+HA(11)
      IF (HA(12).GT.HR1)GO TC 31
```

```
      QUAL1 = (HA(12)-HA(3))/(HR1-HA(3))
      SA(12) = SA(3)+QUAL1*(SR1-SA(3))
      V(12) = V(3)+QUAL1*(VR1-V(3))
      T(12) = T(3)
      GO TO 32
   30 KR = 0
      CALL WASP (5,1,T(12),P(12),D,HA(12),KR)
      HA(12) = EFFST*(HA(12)-HA(11))+HA(11)
   31 KR = 0
      CALL WASP (4,2,T(12),P(12),D,HA(12),KR)
      T(12) = T(12)-459.67
      SA(12) = S
      V(12) = 1./D
      QUAL1 = 1.00
C
C     MASS FLOW FRACTION
   32 MDOT = (HA(11)-HA(5))/(HA(11)-HA(4))
      MDOT1 = MDOT*(HA(13)-HA(3))/(HA(13)-HA(2))
      MDOT2 = MDOT-MDOT1
      MDOT3 = 1.0-MDOT
C
      WRITE(6,50)
   50 FORMAT ('1','   P-PSIA       T-F        H-BTU/LBM   V-FT3/LBM    S-BT
     1U/LBM-R')
      DO 60 I = 1,13
      WRITE (6,55) P(I),T(I), HA(I), V(I), SA(I)
   55 FORMAT (1H0, F10.3,F10.2,F12.2,2F12.5)
   60 CONTINUE
      GO TO 20
   90 WRITE (6,80)
   80 FORMAT('0','STEAMTURBINE EXIT IN SUPERHEAT REGION')
      GO TO 7
C
C     CALCULATE WORK AND HEAT TRANSFER
   20 WGT = HA(9)-HA(10)
      WST = MDOT*(HA(11)-HA(12))+MDOT1*(HA(12)-HA(13))
```

```
      WC  = HA(7)-HA(5)
      WP1 = MDOT1*(HA(2)-HA(1))
      WP2 = MDOT*(HA(4)-HA(3))
      WP = WP1+WP2
      WNET = WGT+WSI-WC-WP
      QIN = HA(9)-HA(8)
      FIELDE = WNET/QIN
      QREJ=(HA(13)-HA(1))*MDOT1
      QREG= HA(6)-HA(7)
C
      WRITE (6,83) WGT
   83 FORMAT ('0','WGT =  ',F8.2,'  BTU/LBM')
      WRITE (6,84) WST
   84 FORMAT ('C','WST =  ',F8.2,'  BTU/LBM')
      WRITE (6,85) WC
   85 FORMAT ('0','WC  =  ',F8.2,'  BTU/LBM')
      WRITE (6,86) WP
   86 FORMAT ('0','WP  =  ',F8.2,'  BTU/LBM')
      WRITE (6,87) WNET
   87 FORMAT ('C','WNET =  ',F8.2,'  BTU/LBM')
      WRITE (6,88) QIN
   88 FORMAT ('0','QIN =  ',F8.2,'  BTU/LBM')
      WRITE (6,89) QREJ
   89 FORMAT ('0','QREJ = ',F8.2,'  BTU/LBM')
      WRITE (6,71) QREG
   71 FORMAT ('0','QREG = ',F8.2,'  BTU/LBM')
      WRITE (6,72) MDOT
   72 FORMAT ('C','MDOT =  ',F8.5)
      WRITE (6,73) MDOT1
   73 FORMAT ('C','MDOT1 = ',F8.5)
      WRITE (6,74) MDOT2
   74 FORMAT ('C','MDOT2 = ',F8.5)
      WRITE (6,75) MDOT5
   75 FORMAT ('C','MDOT5 = ',F8.5)
      WRITE (6,76) QUAL
   76 FORMAT ('C','QUAL =  ',F8.5)
```

```
   WRITE (6,77) QUAL1
77 FORMAT ('0','QUAL1 = ',F8.5)
   WRITE (6,91) FIELDE
91 FORMAT ('0','FIELDE =  ',F8.5)
 7 DO 11 I = 5,16
11 T(I) = 0.0
   GO TO 5
70 STOP
   END
```

Appendix B. Regenerator and Heater Size Calculations

The size of the regenerator and heater determine the cost. The heat transfer area is calculated by using the equation $q = UA\Delta T_{\ell m}$ where

q = heat transfer rate, BTU/hr.
U = overall heat transfer coefficient, BTU/hr ft^2 ^oF
A = heat transfer area, ft^2
$\Delta T_{\ell m}$ = log mean temperature difference, ^oF

The overall heat transfer rate is a function of the heat transfer coefficients of the hot and cold fluids and the thermal conductivity of the tube materials. The heat transfer coefficients for the steam can be determined from the equation

$$\frac{hD}{k} = .023\left(\frac{GD}{\mu}\right)^{.8}(Pr)^{.4}$$ from reference 18 for the steam in the tubes and

$$h = \frac{kC}{D}\left(\frac{D_oG}{\mu}\right)^{.6}(Pr)^{1/3}$$ for the flow outside of the tubes. The regenerator is for the cycle of 200 psia base pressure and pressure ratio of 4. The high pressure steam is assumed to be flowing inside a two inch tube. Steam properties are from the computer program results and ASME Steam Tables.

Data: $Th_1 = 822^oF$ $\quad Th_2 = 559^oF$ $\quad$ Heat Transfer 135.72 BTU/lbm
$Tc_1 = 530^oF$ $\quad Tc_2 = 714^oF$
Low Pressure = 200 psia $\quad$ Power Output = 122.50 BTU/lbm
High Pressure = 800 psia

For the steam in the tubes:

$$h = 387 \text{ BTU/hr ft}^2 \, ^oF$$

For the steam outside of the tubes:

$$h = 108 \text{ BTU/hr ft}^2 \, ^oF$$

The overall heat transfer coefficient is:

$$U = 80 \text{ BTU/hr ft}^2 \, ^oF$$

The heat transfer area, A, for a 1000 MW plant is calculated as:

$$A = \underline{7.89 \times 10^5 \text{ ft}^2}$$

The size of the heater is estimated using data from reference 10 for the gas heat transfer coefficients and temperatures.

Data: Inlet steam temperature = 714^oF
Outlet steam temperature = 1200^oF
Enthalpy Difference = 276.98 BTU/lbm
Inlet Gas temperature = 1915^oF
Exit Gas temperature = 1580^oF
Heat transfer coefficient of gas = 10 BTU/hr $ft^2 {}^oF$

For the steam flow in the tube:

$$h = 343 \text{ BTU/hr ft}^2 {}^oF$$

$$U = 10 \text{ BTU/hr ft}^2 {}^oF$$

$$\Delta T_{lm} = 789^oF$$

$$A = 9.78 \times 10^5 \text{ ft}^2$$

The costs of the regenerator and heater are based on heat exchanger costs from the General Electric study [19]. The cost per square foot for heat exchanges operating at temperature less than 800^oF was \$30. For those over 800^oF the cost was \$200 per square foot. Using these costs adjusted to 1970 dollars the regenerator and heater cost per kilowatt are:

Regenerator - \$18/KW

Heater - \$144/KW

Thesis 165496
M26945 McPherron
An evaluation of the
gaseous-steam cycle
(field cycle).